Atomic Energy Control Board
P.O. Box 1046
Ottawa, Canada
K1P 5S9

Commission de contrôle de l'énergie atomique
C.P. 1046
Ottawa, Canada
K1P 5S9

INFO-0168

PRIMARY PUMP VIBRATION UNDER ACCIDENT CONDITIONS

by

Davis Engineering Limited

A report prepared for the
Atomic Energy Control Board
Ottawa, Canada

June 1984

Canada

Research report

86-02010/

Les opinions exprimées dans ce rapport sont celles des auteurs et n'engagent pas la Commission de contrôle de l'énergie atomique.

The opinions expressed in this report are those of the authors and do not necessarily reflect those of the Atomic Energy Control Board.

Copy Number:
Issue: 3
Date: 18/06/84

PRIMARY PUMP VIBRATION UNDER ACCIDENT CONDITIONS

Prepared By: ....
B.M. Guthrie, P.Eng.,
Project Engineer

...Tom...Currie......
T.C. Currie, P.Eng.,
Subcontractor

Reviewed By: ....
A.J. Bosik, P.Eng.,
Project Manager

Approved By: ....
W.R. Davis, P.Eng.,
President

PRIMARY PUMP VIBRATION
UNDER ACCIDENT CONDITIONS

This work was produced by Davis Engineering Limited under contract to the Atomic Energy Control Board, Contract Serial Number OSQ83-00213.

ABSTRACT

This report presents the results of an international survey on the subject of vibration in nuclear primary coolant pumps due to two-phase flow, accident conditions.

Substantial effort was spent in establishing contacts in the United States and Europe, with insignificant results. Data from potential relevant international references was neither obtained nor analyzed due to time and cost restraints. However, many partially-related operating incidents, test programs and analyses have been referenced.

The literature search also revealed few Canadian references other than those of Ontario Hydro. A substantial amount of work performed by Ontario Hydro has been retained and reviewed.

Ontario Hydro's work has been extensive. Confidence in the mechanical integrity of the pumpsets is good, given the extent of the testing. However, conclusions with respect to piping integrity and thermal-hydraulic performance are difficult to determine due to the inexact geometry of the piping and the difficulties in estimating fluid conditions at the pump. The tests help to understand the phenomena and provide background information for analysis, but should be applied with caution to plant analyses.

The analyses of Ontario Hydro are based on existing theory. However that theory contains many (recognized) assumptions. Some of the perceived limitations in Ontario Hydro's work include (i) the disagreement of empirical and software results, (ii) use of temperature only in analyzing some data, (iii) the lack of data on low speed testing, (iv) the limited blowdown/refill capability of the loops, (v) the limitation on testing due to piping/hairpin vibration, (vi) limitations in the data acquisition and reduction systems, and (vii) the assumption for plant analyses of piping excitation in the first mode only.

Much of the discussion in the report relates to pump head instability. This is perceived to be the most important flow regime causing vibration, as attested by the emphasis of the reviewed literature.

A method for quantitative assessment of the forcing functions acting on the pump-piping system due to void generation and collapse is recommended. A relatively fundamental analytical approach is proposed, supplemented by reduced scale testing in the latter stages.

RÉSUMÉ

Le présent rapport fait état des résultats d'une étude internationale sur les vibrations des pompes de caloportage primaire dues à un écoulement déphasé dans des conditions d'accident.

Des efforts importants ont été consacrés à établir des contacts tant aux États-Unis qu'en Europe, mais n'ont pas porté fruit. Bien que des données provenant de sources internationales pertinentes possibles n'aient pu être obtenues ou analysées à cause de restrictions de temps et de coût, plusieurs incidents, programmes d'essai et analyses partiellement reliés ont été documentés.

La recherche documentaire a également révélé qu'il existait peu de sources canadiennes, à part celles d'Ontario Hydro dont une part importante a été retenue et étudiée.

Le travail d'Ontario Hydro est approfondi. L'intégrité des assemblages de pompes est fiable d'après l'étendue actuelle des essais. Toutefois, les conclusions concernant l'intégrité des canalisations et le rendement thermo-hydraulique sont difficiles à determiner à cause de la disposition géométrique non représentative des canalisations et des difficultés à déterminer les conditions du fluide dans la pompe. Les essais ont aidé à comprendre les

phénomènes et à procurer des renseignements généraux pour analyse, mais devraient être appliqués avec précaution aux analyses de centrales.

Les analyses d'Ontario Hydro sont basées sur la théorie actuelle qui comprend, néanmoins, plusieurs suppositions (reconnues). Parmi les limitations observées dans le travail d'Ontario Hydro, il faut compter (i) les différences entre les résultats empiriques et informatiques, (ii) le recours à la température comme facteur unique dans l'analyse de certaines données, (iii) le manque de données pour les essais à faible vitesse, (iv) la capacité limitée des boucles à se décompresser et à se remplir, (v) la limitation des essais à cause des vibrations dans les canalisations et les sections courbées en épingle à cheveux, (vi) les limitations des systèmes de saisie et de réduction des données, et (vii) l'excitation des canalisations dans le premier mode seulement, pour les analyses de centrales.

Une grande partie de la discussion du rapport traite de l'instabilité dans les têtes de pompe qui est considéré comme le facteur d'écoulement le plus susceptible de causer des vibrations, d'après l'importance qu'y accorde la documentation examinée.

Le rapport recommande une méthode d'évaluation quantitative des fonctions de force agissant sur le système de pompes et de canalisations, dues à la production et à la suppression du vide. Il présente enfin une approche d'analyse relativement élémentaire à laquelle s'ajoutent des essais à échelle réduite pour les dernières étapes.

PREFACE

The objective of this project is to review the state-of-the-art of two-phase flow induced vibration in nuclear primary coolant pumps. The primary tasks are as follows:

(1) search Canadian and international literature and information;

(2) critically review the relevant information obtained; and,

(3) recommend approaches to analyzing the subject phenomenon.

The specific technical areas of relevance include the following:

- detailed observations from site incidents and experiments;
- mechanisms of the pipe/flow oscillations induced by void generation and collapse near the pump suction;
- the effect on the pump/piping vibration of system parameters; and,
- analytical and empirical methods developed to correlate pump-piping vibrations to system parameters.

The pump vibration/two-phase flow problem itself has not been analyzed; the relevant literature obtained has been reviewed critically and an approach for analyzing the problem has been recommended.

The statement of work for this contract included a search and review of Canadian and international references. The literature search revealed few relevant Canadian references other than those of Ontario Hydro. As for the international references, a substantial effort was spent in establishing contacts in the United States and Europe. Unfortunately data from several potentially relevant international sources was neither obtained nor analyzed due to time and cost restraints of this contract.

The report contains few references to (i) forcing mechanisms other than head instabilities, (ii) slug or intermittent flow regimes, and (iii) internal pump fluid/structure interaction, due to the lack of relevant analyses in the reviewed literature.

TABLE OF CONTENTS

PAGE

ABSTRACT

PREFACE

A. EXECUTIVE SUMMARY 1

B. BACKGROUND 3
  1. CANDU Designs 3
  2. Accident Scenarios 3

C. LITERATURE SEARCH 5

D. OPERATING EXPERIENCE 7
  1. Operating Incidents 7
  2. Test Facilities 13

E. PIPE/FLOW OSCILLATIONS 18
  1. Bruce/SLOCAT A Tests 18
  2. Bruce SLOCAT B Tests 22
  3. Pickering B Tests 24
  4. Air-Water Tests of Rothe et. al....................... 26

F. EFFECT OF SYSTEM PARAMETERS 29
  1. Flowrate 29
  2. Pressure/Temperature 29
  3. Void 30
  4. Voiding Rate 31

G. ANALYTICAL AND EMPIRICAL MODELS............................. 32
  1. Analytic Models....................................... 32
  2. Pump-Induced Flow Oscillations........................ 33
  3. Empirical Models...................................... 35

PAGE

H. RECOMMENDED METHODOLOGY 37

I. CONCLUSIONS AND RECOMMENDATIONS 40
  1. Conclusions 40
  2. Recommendations 42
  3. Limitations 42

REFERENCES 44

ACKNOWLEDGEMENTS 49

TABLES 50

FIGURES 53

APPENDICES

LIST OF ACTIVITIES, SEARCHES, CONTACTS..........APPENDIX A
PUMP DESIGN DATA ....APPENDIX B
VISIT REPORT - ONTARIO HYDRO..........APPENDIX C
SELECTED BIBLIOGRAPHY..........APPENDIX D
TELEX MESSAGES..........APPENDIX E

TABLES

PAGE

TABLE 1 Calculated Natural Frequencies, Allowable Deflections for the Pump Test Loop Model Assuming No Water in the System, and Observed Deflections, Bruce NGS A Pump Tests 50

TABLE 2 Maximum Vibration Amplitudes Measured at Different Flowrates in the Bruce NGS A Pump Tests 51

TABLE 3 Summary of Pickering NGS B PHT Pumpset Simulated Pickering NGS B PHT Pumpset 52

FIGURES

PAGE

FIGURE 1 PHT Pump Test Loop - Flow Diagram - Bruce Loop 53

FIGURE 2 SLOCAT Pump Head and Pipe Vibration vs Time, Test 69, Bruce GS A 54

FIGURE 3 SLOCAT, Pump Head and Pump Vibration vs Time 71, Bruce GS A 55

FIGURE 4 Oscillogram of Discharge and Suction Pressure, Venturi Head, Hairpin Displacements, and Shaft Runouts ..., SLOCAT Test 85, Bruce GS A 56

FIGURE 5 Pictorial Presentation - Pump Head Variation vs Void - Showing Areas of Instability - Bruce GS B 57

FIGURE 6 Pickering B PHT Pump Test Loop Flow Schematic Diagram 58

FIGURE 7 Test 97 (250°C), Pickering NGS B Test Loop 59

FIGURE 8 Test 202 (100°C), Pickering NGS B Test Loop 60

FIGURE 9 Pressure Pulse Amplitude Verses Matrix Test Temperature, Pickering NGS B Test Loop 61

FIGURE 10 Test 233 (50°C), Pickering NGS B Test Loop 62

FIGURE 11 Schematic of Test Facility, Rothe, et al. 63

FIGURE 12 Idealized Pump Surge System, Rothe, et al. 64

FIGURE 13 Crossplot of Pump Performance Data from Air-Water Tests 65

A. EXECUTIVE SUMMARY

This project was initiated as an international survey of the state-of-the-art on the subject of pump vibration under two-phase flow conditions. The initial search revealed few relevant references. Further pursuit of the international contacts, and a review of the relevant references (largely those of Ontario Hydro) followed. Thus, this report reflects (i) an international search, and (ii) an overview of Ontario Hydro's work.

Although many potentially relevant references have not been obtained, it can be stated with some confidence that Ontario Hydro has carried out the greatest amount of work in the world in the subject matter. Many other references have done semi-relevant work some may even have vibration data, but their objectives have not been oriented to the same question, and the effort to digest and relate their work is beyond the scope of this project. The operating incidents and empirical studies summarized in Section D should be pursued if further investigation or analyses are to be considered.

Ontario Hydro's work has been extensive. Confidence in the mechanical integrity of the pumpsets is high, given the extent of the testing. However, conclusions with respect to piping integrity and thermal-hydraulic performance are difficult to determine due to the inexact geometry of the piping and the difficulties in estimating fluid conditions at the pump. The tests help to understand the phenomena and provide background information for analysis, but should be applied with caution to plant analyses. The analyses of Ontario Hydro are based on existing theory. However that theory contains many (recognized) assumptions which could be refined in future work.

Some specific perceived limitations to Ontario Hydro's work are addressed below:

(1) dissimilarity of piping and supports of the test rigs versus the plants (including single versus multiple loops);

(2) disagreement between the SOPHT computer code results and empirical results;

(3) difficulties in measuring pump suction conditions, especially void;

(4) correlation of vibration and other paramters against temperature only;

(5) the lack of data at high voiding rates;

(6) the lack of results or limited range of data for many tests, due to: excessive vibration (Bruce loop hairpins), limited blowdown/refill capability (Bruce loop), limitations in the data acquisition and reduction systems, problems in low speed testing; and,

(7) the assumption (for plant analyses) that piping is excited in the first mode only.

Much of the discussion in the report relates to pump head instability. This is perceived to be the most important flow regime causing vibration, as attested by the emphasis of the reviewed literature.

The recommended methodology (for quantitative assessment of the forcing functions acting on the pump-piping system due to void generation and collapse) is necessarily very general, investigating the following (in sequence):

(1) the uncoupled fluid/structural assumption;

(2) steady-state, two-phase pump performance characteristics;

(3) a hydro-acoustic model of the loop;

(4) the piping forcing function;

(5) piping vibration, using a finite-element program; and,

(6) a scaled test facility (in parallel with steps (3), (4) and (5)).

B. BACKGROUND

B.1 CANDU Designs

Operational, experimental and analytical results from numerous sources have been searched. However, in evaluating vibration models or proposed methodologies for analysis, the specific characteristics of the particular pump/motor set are critical. For this reason four CANDU designs have been labelled the base cases:

(1) Pickering NGS B
(2) Bruce NGS B
(3) Darlington NGS A
(4) Point Lepreau/Gentilly-2

Details and figures of these pumpsets are provided in Appendix B as background information.

B.2 Accident Scenarios

The loss of coolant accidents (LOCA's) of interest are characterized below in order of decreasing priority:

(1) Large LOCA with emergency coolant injection, ECI (100% break of pump suction header)

- rapid drop in pressure;
- voided conditions at lower temperatures (100-130°C);
- refill at low pressure (slightly greater than saturation) and temperature (100°C);
- slug flow in pump.

(2) Very small LOCA without ECI (0.375% break of reactor inlet header, inner zone)

- system voiding and pressure drop;
- gradual cooling;
- extended running of pump in cavitation conditions.

(3) Other scenarios (between the above two extremes) which may create pump vibration problems.

The above transients are dynamic; the thermal-hydraulic conditions at the pump change with time and are very dependant on conditions in the reactor and the rest of the Primary Heat Transport System. For this study a control volume approach has been followed as much as possible; i.e. only conditions just upstream and downstream of the pump are considered.

In order to further limit the scope of the study, accident scenarios* which may be indirectly related to two-phase flow conditions are not considered. (However the seismic design of pumps is partially relevant.)

Complex loop dynamics can occur during a LOCA, as characterized by two example descriptions:

- At Bruce B, primary coolant makeup can be provided by normal outsurge from the pressurizer, feed from the D_2O storage tank, or the Emergency Coolant Injection system (0.45 m^3/s).
- The Darlington ECI system has 8 injection points, at the pump discharges and at the steam generator inlets. Interconnections between the two loops help to provide makeup water to the troubled loop before interconnecting valves close (after initial injection).

---

* Such as - earthquakes, hydrogen explosions, steam/water environments in containment, electrical failures, random pump component failures.

C. LITERATURE SEARCH

A list of activities, searches and personal contacts is given in Appendix A. A selected bibliography is provided in Appendix D. (Note that Appendix D is different from the References.)

Of all information obtained, the work of Ontario Hydro is by far the most relevant. The technical sections of this report are based largely on Ontario Hydro documents.

The failure to unearth relevant international references can be attributed to several factors:

- directly relevant work is not being done;
- time and cost restraints of this contract prohibit a more complete search and the timely retention of semi-relevant documents;
- semi-relevant work is underway and results are not yet available; and,
- directly relevant work is being done, but is not publicly available, for commercial proprietary reasons.

A significant amount of work has been directed to semi-relevant fields such as "two-phase flow in pumps", "pump vibration", and "flow-induced vibration in pipes". (Most of the references in Appendix D can be classified as semi-relevant). Very few references specifically address "two-phase flow induced vibration in primary coolant pumping systems".

Interesting operating incidents and test programs are addressed in Section D, which follows. In some cases written information was not obtained. In virtually all cases, the level of effort required to analyze the large number of semi-relevant references and relate them coherently to the specific question at hand is beyond the scope of this project.

Several summary statements can be presented from the search:

(1) Most contacts have little relevant information, but they usually know of other "potential" contacts (many of whom cannot be reached).

(2) Virtually all sources who are familiar with the question technically cite Ontario Hydro's work as the state-of-the-art.

(3) Most North American contacts have very little idea of what was going on in the rest of the world.

(4) Pump manufacturers have generally not analyzed the technical problem because it is beyond their scope. (Alternatively, they consider relevant analyses or tests confidential).

(5) The regulatory and technical climate in the United States is different from the rest of the world. "While the United States utilities are in the process of determining safe minimum flow limits for nuclear applications, the European utilities specify proof that the same pumps can withstand a severe "dry-run" (vapour locked) operating mode without any visual damage to pump internal components. Such requirements would be quite unique in the United States today". (Reference (2), E. Makay's Discussion). Also, Light Water Reactor designs are more conducive to thermosphoning than CANDU's.

(6) The Europeans are perceived to be more involved in vibration (but not necessarily two-phase flow) than North Americans (Reference 3).

(7) Previous literature reviews have reached similar, non-conclusive results (References 3 and 4).

D. OPERATING EXPERIENCE

D.1 Operating Incidents

Several incidents have been cited as relevant or possibly relevant, and are described very briefly below. Further information may be forthcoming from the Organization for Economic Co-operation and Development (OECD), Incident Reporting System in Paris, and other sources.

In general, the data from operating stations is insufficient to make conclusive analyses. This is due to the limited instrumentation, and the time limitations and priorities of operators during such incidents. Even if data are recorded, conclusions can be difficult or conflicting, depending on the depth of any subsequent analyses (for example, Reference 5). For these reasons, system thermal-hydraulic transients cannot be presented in detail here.

D.1.1 Bruce NGS "A", Unit 2, August 16, 1975

(initial condition)	- commissioning test;
(time 0:00 seconds)	- liquid relief valve opening; - rapid drop in heat transport pressure; - makeup from D_2O storage tank;
(time 0.04 seconds)	- drop in gland supply pressure;
(time 0.11 seconds)	- main coolant pumps cavitate; - estimated 116^{0}C primary heat transport (PHT) system temperature, 2.5% loop average void; (see Notes); - severe water hammer and noise, some wall vibration; - pump discharge pressure bounces erratically at times off-scale (15 MPa); - pump motor current bounces erratically;

(time 3.32 seconds)

- pumps switched off and braked;
- heat transport pressure continues to drop;

(postscript)

- hangers and supports damaged;
- pumpset running well after 5 years.

Notes:

Initially, Ontario Hydro's Nuclear Operations implied that there was only cavitation in the impeller blades, caused by insufficient net positive suction head (NPSH). Citing the Pickering A tests, the Nuclear Systems Department (NSD) suggested that, because there was pipe vibration, there was two-phase instability (a non-solid system), corresponding to a minimum 2.5 percent loop average void. NSD checked this assumption with Bruce B (commissioning) flow calculations and the SOPHT thermal-hydraulics code. Furthermore, NSD's volumetric calculations gave rise to the conclusion that the system could have been voided, probably at a level less that 0.5 percent.(Ref. 5).

Given that (i) there were pressure spikes, (ii) conditions at the pump are localized and (iii) pressures at the pump suction are the lowest in the system, NSD's contention that there was voiding seems reasonable. However, substantiation using Pickering test loop results, SOPHT predictions and estimates of local conditions from total system volumetric calculations seems tenuous.

D.1.2 Pickering NGS "A", Unit 1, May 11, 1978 (Ref. 6)

- significant event (SER #78-61)

(time 0:00 hours)

- full reactor power;
- relief valve fails open;
- rapid drop in heat transport pressure (to 5.5 MPa);

(time 0:04)

- loss of coolant trip;
- partial moderater dump;
- manual closure of valve;

(time 0:13)

- steam reject valves slowly opened manually, to increase cool-down rate;
- constant decay of heat transport pressure;
- manual trip of turbine;

(time 0:23)

- failed relief valve closed manually;
- HT pressure 2.7 to 3.4 MPa; temperature 199°C;

(time 0:29)

- HT pressure still decaying;
- pumps cavitating intermittently;

(time 0:34)

- south loop pumps still cavitating; are shut down;
- north loop pressure control re-established;

(time 0:52)

- south loop pressure restored (reached low pressure of 1.37 MPa);
- pumps restarted;

(time 1:04)

- poison outage still in effect;
- system stabilizes at 2.75 MPa, 149°C

D.1.3 Douglas Point G.S., August 28, 1975 (Ref. 7)

(initial condition)

- significant event (SER #75-20);
- 2% full power, HT temperature 350°F;

(time 0:00 hours)

- high vibration reading on primary pumps (0.36 in/sec);
- high gland recirculating temperature (after 10 seconds);
- no changes in flow or pressure;

(time 0:01)

- vibration exceeds 0.6 in/sec;
- pump shutdown;

(time 0:01:30)

- drop in vibration level;
- pump restart;
- high gland recirculating temperature (after 3 seconds);
- high vibrations (>0.6 in/sec) and currents in adjacent pumps;
- high vibration (>2 in/sec); rise in motor current to 170 amps; pump shutdown;
- adjacent pumps return to normal;

(postscript)

- pump appeared to have seized; taken out of service for inspection.

D.1.4 Three Mile Island, Unit 2, Pennsylvania (Ref. 8)

(initial condition)

- normal operating power;

(time 0:00 min:sec)

- interruption of feedwater flow to steam generators;
- overpressure in reactor coolant system;
- relief valve opening;
- reactor shutdown;
- reduction in system pressure (to normal levels);

(time 0:40)

- relief valve fails to close;
- loss of coolant, (continues for 2.4 hours);
- decrease in system pressure;

(time 2:00)

- high-pressure water injection into reactor coolant system;

(time 3:00)

- high-pressure injection stopped by operator;
- net loss of coolant;
- steam voids outside pressurizer and in loop continually increase;

(time --> 73:00)	- main coolant pumps remain on;
(time 73:00)	- indications of low system pressure, high vibration, low flow; - pumps in loop B shut off; - steam and water separation in loop B;
(time 100:00)	- loop A pumps shut-off (for same reasons as above); - indications of separated steam and water mixture in loop;
(time : 1.9 hours)	- rapid rise in reactor coolant outlet temperature; greater than 620°C after 2.5 hours; - reactor coolant outlet piping remains for 10 hours in superheated steam region (plus non-condensible gas);
(time : 2.4)	- block valve in series with relief valve closed; - operators try to re-establish cooling for next 13 hours: by natural & forced circulation through steam generators, or decay heat removal system; - actuation of high pressure injection; - circulation not possible due to non-condensible gas; - wide variations in system pressure;
(time : 7.5)	- hydrogen vented during depressurization; - actuation of decay heat removal system, but pressures still too high;
(time : 11.5)	- closure of block/relief valve; - low rate of coolant injection; - steam generators still isolated;
(time : 13.5)	- sustained high pressure injection; - system repressurization; - some hydrogen still in steam generators; - re-establishment of forced circulation via pumps, and cooling through one steam generator.

Note: The pumpsets in the TMI accident are generally conceded to have seen a wide range of thermal-hydraulic and vibration conditions. They remained intact, and were useful to the operators in efforts to control the loop conditions. This shows that the pumpsets can be dependable under certain extended "small-break" incidents. Additional measurements of vibration and void would aid operators in monitoring pump integrity under such conditions.

D.1.5 Surry, Unit 1, Virginia, November 30, 1973 (Ref. 9)

- excessive vibration of reactor coolant pump;
- fluctuations in primary coolant flow;
- station taken off line.

D.1.6 Caorso Power Station, Italy, 1981 (Ref. 10)

- previous oscillations of recirculating pump motor current;
- full power operation;
- noise in RHR system line connected to reactor recirculation system unidentified;
- no corresponding variations in thermal-hydraulic parameters;
- follow up tests (July 1981);
- no vibrations dangerous to plant structures;
- motor current oscillations reduced by improving regulation system/recirculation control system.

D.1.7 Dampierre, Unit 1, France, February 29, 1980 (Ref. 11)

- unit being prepared to go critical;
- valve failure open;
- primary circuit pressure dropped to 0.5 MPa;
- damage to No. 1 seal of the three primary pumps;
- damage to No. 3 seal of one primary pump.

D.1.8 Fukushima Daini, Unit 1, Japan, November 19, 1981 (Ref. 12)

- pre-operational test;
- trip of 2 turbine-driven reactor feed pumps;
- reactor scramm on low water level;
- cavitation in low pressure condensate pump due to valve falsely open.

D.2 Test Facilities

There are many pump test facilities in the world, but most of them have been built to analyze thermal-hydraulic or mechanical design aspects. Very few have recorded vibration data or even qualitative statements regarding vibration. Some of the relevant and semi-relevant facilities are addressed below.

D.2.1 Ontario Hydro - Kipling Pump Test Facility, Toronto

- general results discussed in body of this report (Sections E, F, and G).

D.2.2 Westinghouse, Hamilton, Ontario (Ref. 13, 14)

- two major projects of relevance:
  (1) LOBI tests;
  (2) RD-14;
- LOBI tests:
  - small model (100 hp) of European pressurized water reactor (PWR) pump;
  - hydraulic, torque measurement data; sent to Ispra Establishment, Italy;
  - vibration data or judgements not available;
  - cavitation a small part of the work;
  - two reports on order from Atomic Energy of Canada Ltd. (AECL),

Chalk River Nuclear Laboratories (CRNL)

- RD-14
  - sponsored by Ontario Hydro and AECL, Whiteshall Nuclear Research Establishment (WNRE);
  - work still underway;
  - final report should be available in June (through Hydro and WRNE);
  - some vibration measurements, but largely a thermal-hydraulic study; i.e. no vibration discussion.

D.2.3 Thermal-Hydraulic Studies, Electric Power Research Institute (EPRI)-Sponsored (Ref. 15)

- includes several experimental test rigs:

  (1) Aerojet Nuclear Company (ANC), Idaho;
  (2) Babcock & Wilcox, Virginia;
  (3) Combustion Engineering, Connecticut;
  (4) Creare, New Hampshire;
  (5) Massachusetts Institute of Technology (MIT), Massachusetts;
  (6) Davis Engineering, Ottawa;

- all tests were strictly thermal-hydraulic oriented; no vibration data available with the exception of Combustion Engineering which may have raw vibration data available (Ref. 16,17,18).

D.2.4 Loss of Fluid Test (LOFT) Facility, Idaho (Ref. 19,20)

- focussed on thermal-hydraulics only;

- have some raw vibration data, not assembled or analyzed;

- pump, motor and piping are very different, (i.e. thrust bearings, "wet" motor type, anchoring).

D.2.5 Byron-Jackson Pump Div/Borg Warner, Los Angeles, California (Ref. 21)

- sponsored by Southern California Edison Co. as a seal integrity test under loss of component cooling water;

- survey of utilities showed one operating incident where a similar pump ran in excess of 30 minutes after loss of component cooling water;

- separate tests on motor and pump showed some wear on the seals;

- shaft vibration was monitored at lower ends of spacer coupling; not a problem;

- no other vibration or thermal-hydraulic data.

D.2.6 CETIM, France (Ref. 22)

- unsteady flow through a small centrifugal pump;

- thermal-hydraulics, no vibration or specific two-phase flow data;

- little information available.

D.2.7 Electricite de France, Chatou, France (Ref. 23-26)

- two-phase thermal-hydraulic performance of 2 MWe centrifugal pump; (INDIRA/EPOPEE);

- no vibration data;

- little information available; no response to a telex.

D.2.8 CEA - Centre d'Etudes Nucleares de Saclay, Gif-sur-Yvette, France (Ref. 27,28)

- tests on flow induced vibrations in piping;

- **GASCOGNE** loop at D.E.M.T.; **SAFRAN** loop;
- acoustic excitations of tube and elbows;
- comparison of result to simulations (VIBRAPHONE acoustic program);
- little information available, no response to a telex.

D.2.9 Kraftwerk Union, Karlstein Labs, Germany (Ref. 29,30)

- full scale test of boiling water reactor (BWR) internal recirculation pumps under cavitation and vibration conditions;
- axial and half-axial impellers;
- very different geometry from CANDU systems;
- 50-279°C temperature range; various rotative speeds;
- shaft vibration measurements;
- some results discussed in Section F.2.

D.2.10 LOBI Project, Ispra Establishment, Italy (Ref. 31)

- influence of two-phase flow conditions on pump head and torque investigated;
- no vibration testing.

D.2.11 CISE Labs, Milan, Italy (Ref. 32)

- relevant experimental research using test loop scheduled to start in the next year.

D.2.12 Turin Polytechnic, Turin, Italy (Ref. 32)

- relevant experimental research using test loop scheduled to start in the next year.

D.2.13 Polish Academy of Sciences, Gdansk, Poland (Ref. 33)

- cavitation in mixed flow pump with an open shroud impeller;

- non nuclear-specific;

- measurements of hydraulics, noise at impeller, pump body vibration, pressure pulse spectrum, etc;

- very little data presented.

D.2.14 Technical University of Budapest, Hungary (Ref. 34)

- cavitation/vibration tests on centrifugal pumps;

- two test rigs;

- non nuclear-specific;

- little data presented, but interesting discussion.

D.2.15 General Electric, Moss Landing (U.S.)

- no information available.

E. PIPE/FLOW OSCILLATIONS

A survey of the literature related to vibration and flow oscillation problems in two-phase flow loops with centrifugal circulating pumps revealed very few references of relevance to the present study outside of those prepared by Ontario Hydro. Of the reports issued by Ontario Hydro, references 35 and 36 describe the results of tests of a Bruce NGS A primary heat transport (PHT) pump under two-phase conditions in a test loop described in reference 37. References 38 and 39 describe the results of similar tests performed on a Bruce A pump fitted with a Bruce B motor. Tests of a Pickering NGS B PHT pump in a different test loop are described in references 40 and 41. In the only relevant reports not prepared by Ontario Hydro, Rothe et al (Ref. 42) describe the results of experiments performed (in an air - water loop) to study pump surge in two-phase flow, and Schafer and Simon (Ref. 29) briefly relate vibration, void and temperature in a test facility.

Before attempting to identify particular operating conditions under which pump-excited two-phase flow oscillations will or will not occur, the results of the Bruce and Pickering pump tests performed by Ontario Hydro and the observations of Rothe et al (Ref 42) will be summarized briefly.

E.1 Bruce SLOCAT "A" Tests

The performance characteristics and operating conditions (nominal) of the Bruce NGS A PHT pump in the test loop are given below

Operating temperature - $264^{\circ}\mathrm{C}$
Operating pressure - 10.4MPa
Rated flow - $3.3\ \mathrm{m}^3/\mathrm{s}$
Rated head - 190.5 m
Motor rating - 8.5 MW
Motor speed - 1800 rpm
Specific speed - $3220\ \dfrac{\text{rpm US gpm}^{1/2}}{\text{ft}^{3/4}}$

A pump with the nominal performance characteristics noted above was tested in the two-phase flow loop shown in Fig. 1. The test loop did not duplicate exactly the mechanical or hydraulic characteristics of a Bruce A PHT circuit and hence would not be expected to provide flow oscillation data applying directly to the real reactor. The loop was equipped with a flow measuring venturi and pressure transducers along with several accelerometers to measure vibration levels at different points in the piping network. The average loop void was calculated from the holding tank volume, but the void fraction at the pump suction could not be successfully measured (and was therefore estimated).

The initial, primary objective of the Ontario Hydro test program was to investigate the mechanical/structural integrity of the pumps, primarily in single-phase conditions. The subsequent SLOCAT (Small-Break Loss of Coolant Accident Test) programs were initiated after Three-Mile Island, to test the pumps "over a wide range of two-phase flow conditions", to check the operations directive of keeping the pumps running during a LOCA (Ref. 36).

Because of the geometric differences between the test loop and the actual PHT system, and because of the difficulty in measuring the pump suction conditions, the application of test data to real reactor transients seems somewhat tenuous. Thus, these two-phase tests should be used for understanding of the phenomena only.

The only energy source in the loop was the pump itself (i.e. heaters were not used) so a typical test began by running the test pump at full speed (with the loop suction pressure held at 9 MPa) until the desired test temperature was reached. The pump speed was then set to the desired value and water was gradually bled from the loop. When a large average void fraction of typically 50%* was reached (the flow in the loop had ceased by this time), cold water was pumped back into the loop at a rate of 6 l/s by the pressurizing pump. The test was terminated when the pump suction pressure reached 9 MPa. Because of the test procedure adopted, the test temperature varied slightly as the loop was bled and dropped to 100°C or less (typically) during the refill period. Tests of the Bruce A pump were performed at pump speeds of 400 to 1800 rpm (Ref. 36)+. Although it was initially intended to perform tests at temperatures ranging from 265°C to 115°C over the full range of pump speeds just mentioned, full speed (1800 rpm) tests at temperatures below 205°C could not be performed because of excessive vibration of the loop piping (hairpin) during refill. From the measurements of (1) the pipe vibration amplitudes at the north "hairpin", (2) the high and low pump heads, and (3) the frequency of the pump head pulsations (all of which are shown plotted versus time in Figs. 2 and 3), for the tests performed at initial temperatures of 235°C and 215°C respectively at approximately 1800 rpm, it is apparent that the large vibration amplitudes that were observed during refill occurred when the frequency of the head pulsations was 1.5 Hz to 2.2 Hz. Since the fundamental frequency of the test loop piping was approximately 1.5 Hz (Ref. 35) the large vibration amplitudes seen at times of 15:12 - 15:15 in Fig. 3 (when the frequency of the head pulsations was approximately 1.5 Hz) occurred because the head pulsation and piping fundamental frequencies were matched. Figures 2 and 3 show that the head pulsation amplitude was not at its largest value when the highest pipe vibration amplitude was recorded in either test* so the lower frequency of the head pulsations that produced the large vibrations during refill must have been the primary cause of the severity of those vibrations.

---

* This is an expected design critera. In actual tests, an upper limit of 20-22% void fraction was reached.

An oscillogram of several parameters measured during the period when the highest vibrations were recorded in the full speed test performed at 205°C is given in reference 36, and is included here as Fig. 4. Full speed tests were not performed at temperatures below 205°C because of excessive piping

---

+ Results of the low speed tests are not clear in the referenced report. It was mentioned in the report that flow stoppages due to steam blockages caused these tests to be forshortened, as noted below.

From Ref. 36: (i) @ 262°C, 575 rpm - flow stopped at 21% steam void; suction header filled with steam (inadequate NPSH);

(ii) Test No. 8, @ 400 rpm - flow stoppage in hairpins;

(iii) Test No. 81, 550-1300 rpm - intermittent flow stoppages with increase in pump speed.

* Higher head pulsation amplitudes occurred as the pump head was dropping rapidly soon after the start of loop bleeding or when the head was recovering rapidly during refill.

vibration as mentioned previously . The oscillogram shown in Fig. 4 was obtained during loop refill when the loop temperature had dropped to 182°C.

The center chart of the oscillogram shown in Fig. 4 displays, among other things, the pump discharge, suction and differential pressures as functions of time. The top three traces of the center chart are pressures measured at different points in the discharge piping while the two traces directly beneath them show pressures measured in the suction pipe. The oscillogram shows that the pump suction pressure remained fairly constant with time but that the discharge and differential pressures varied periodically with a frequency of approximately 1.5 Hz, as did the in-plane and normal vibrations of the north hairpin (traces 6 and 7). These results show, once again, that the loop vibrations were most severe during refill because the frequency of the head pulsations was matched to the fundamental frequency of the piping. It is somewhat surprising that large vibrations were not observed when the loop was being bled and the pump head was dropping rapidly because Fig. 2 shows that the head pulsation frequency probably passed through the loop fundamental frequency of 1.5 Hz during that period. The head pulsation frequency was approximately 3.4 Hz at 14:34 in test 69 (to which Fig. 2 applies) for instance, and then dropped to approximately 1 Hz at 14:38. A small peak in vibration amplitude occurred at 14:36 - 14:37, however; so it is possible that the head pulsation frequency passed through the critical region around 1.5 Hz too quickly to have much effect. Koehler (Ref. 39) notes that the rate at which the loop was drained in the Bruce A pump tests was usually higher than the rate at which it was refilled so the head pulsation frequencies would probably have changed more rapidly during draining as a result. According to Koehler (Ref.36) the average loop void during the period to which the oscillogram shown in Fig. 4 applies, ranges from 8.5% (estimated) at 14:19 to 1.5% (estimated) at 14:24.

Table 1 (from reference 36) shows calculated natural frequencies for the Bruce NGS A PHT pump test loop. Since the head pulsation frequency which caused the most severe vibrations of the loop piping (ie. 1.5 Hz), was fairly close to both the first and second mode frequencies of 1.44 Hz and 1.55 Hz respectively, it is likely that the head pulsations excited at least those two modes.

Changes to the loop natural frequency (piping changes and guy supports, for example) were considered mid-way through the test program (Ref. 36) in order to relieve some of the restrictions on the testing, and to analyze the resulting thermal-hydraulic response. However, the alterations considered were not implemented for cost reasons.

The tests to which Figs. 2 - 4 apply were performed with the flow control valves in the three hairpins set to give approximately equal flowrates in the hairpins at the pump's design flow of 3.3 m^3/s (i.e., the flow in each hairpin would have been 1.1 m^3/s). Other tests were performed with the flow control valves adjusted to give different flows in the hairpins and different overall flows. As seen in Table 2, (which is essentially Table I in reference 35, retabulated with the tests performed at similar system temperatures, grouped together to separate the effects of flow and system temperature on pipe vibration), pipe vibration amplitudes decreased as the flow increased.

Since the frequency of the head pulsations is reported to have passed through the fundamental loop frequency of 1.5 Hz in each of the tests noted in Table 2, the reductions in vibration amplitude observed at the higher flowrates probably resulted from increased hydraulic damping. Although the vibration amplitudes would also have been reduced if the head pulsation frequency had passed through the loop frequency of 1.5 Hz too quickly (in the tests at the higher flowrates) for the loop vibrations to become fully developed, there is insufficient data to determine whether this actually happened.

The results of the pump tests performed at pump speeds other than the design value (1800 rpm) are not discussed in any detail in references 35 and 36.

E.2 Bruce/SLOCAT "B" Tests (Bruce A Pump Equipped with a Bruce B Motor)

Although the data obtained in the SLOCAT "A" pump tests described previously suggested that vibrations of the Bruce pump test loop would get progressively worse as the loop temperature was lowered, the SLOCAT "B" tests showed that the vibrations were most severe when the loop temperature was approximately 180°C. (Note that the oscillogram shown in Fig. 4 was recorded when the loop temperature was 182°C). The pressure pulsations occurring at the fundamental frequency of the loop piping continued to cause the most severe vibrations even though the amplitudes of the pressure pulsations occuring at other pulsation frequencies were often larger.

According to Burchett (Ref. 38) the four basic types of instabilities shown in Fig. 5 were observed in the SLOCAT "B" pump tests. Since Burchett describes each type of instability in considerable detail, their characteristics will only be summarized here. The four classifications were as follows:

1) Pump head instability - This type of instability was observed at test temperatures above 170°C while the pump head was degrading or recovering. The associated pressure pulsations were periodic with amplitudes as high as 400 kPa and were accompanied by flow oscillations (at the pump discharge) of the same frequency. The frequency of the pressure pulsations increased as the system void decreased. Pump head instabilities caused the most severe pipe vibrations.

2) General Flow Instability - Instabilities of this type were observed at higher void fractions than the pump head instabilities and were probably slug or churn flow - induced.

3) Interrupted Flow Instability - This type of instability was non-periodic and appeared as a large pressure spike (at the pump discharge). The magnitude of the spike was as large as 0.95 MPa in one instance. Interrupted flow instabilities occurred when the flow in the loop had almost ceased.

4) Oscillating Flow Instability - This type of instability occurred after the system had stratified and measurable flow occurred in the discharge pipe only. A plausible explanation of the mechanism responsible for this type of instability is advanced by Burchett in reference 38.

Koehler (Ref. 39) notes that variations in motor power during periods of head pulsation were generally much less (on a percentage basis) than the head variations themselves, presumably because of the inertia of the pump flywheel. Pump torque was apparently the first monitored variable to change when the pump head fluctuated; changes in the head itself and the motor power lagged a fraction of a second behind.

E.3 Pickering B Tests

Performance characteristics and operating conditions for a Pickering B PHT pump in the test loop are given below:

Operating temperature - 249°C
Operating pressure - 8.48 MPa
Rated flow - $0.765\ m^3/s$
Rated head - 143 m
Motor rating - 142 MW
Motor speed - 1800 rpm
Specific speed - $1960\ \frac{\text{rpm US gpm}^{1/2}}{\text{ft}^{3/4}}$

A pump with the nominal performance characteristics noted above was tested in the two-phase flow loop shown in Fig. 6. The piping on the discharge side of the pump up to the tee replicated the geometry of a Pickering B PHT piping system from the pump to the reactor inlet header. The suction tee was also designed to resemble the piping in the real reactor but the mechanical and hydraulic characteristics of the test loop and a Pickering B PHT system were otherwise different. The loop was equipped with temperature, pressure, flow and vibration sensors as shown in Fig. 6 and, as in the case of the Bruce pump test loop, was instrumented to measure the average loop void. (Measurement of the void fraction at the pump suction was not available.)

The procedure followed in each test was essentially the same as that already described for the Bruce pump tests. A summary of the test conditions examined in the Pickering test program is given in Table 3 (from reference 40). The pump was not operated during the refill period in all of the tests because the void/temperature conditions experienced during refill would have been essentially the same in each case.

Test 194 was one of the tests in which the pump was operated during the refill period. The initial temperature in the test was 250°C (this was the highest starting temperature used) and the initial flow was 100% of the design value. The description of test 194 given in reference 40 does not mention the observation of high vibrations during draining. However, Fig. 7 (from reference 41, test 197), shows a number of system parameters, such as pump differential pressure, pump power and average void, (as a function of time) on which a period of high vibrations is indicated (average void of around 2-7%). The initial conditions for test 197 were the same as those already mentioned for test 194 (ie, 250°C, 100% design flow) so variations in the head, power, etc. measured during the bleeding period were probably quite similar for the two tests (unless different rates of bleeding were used). Note that the high vibration period indicated in Fig. 7 spans the period in which the pump head was degrading rapidly (the pump head curve shown in Fig. 7 does not show rapid oscillations because head values used to plot the curve were 12-second averages, as perceived shortcoming of the data acquisition system).

Similar plots of head versus time given in reference 41, for initial test temperatures ranging from 50°C - 250°C (100% design flow in all cases), show that the highest vibrations observed in each test during the draining period always occurred while the pump head was degrading rapidly. This is seen in Fig. 8, for instance, which shows the head, power, void, etc. plotted versus time for the lowest temperature (50°C) test (test 233). On the basis of the information displayed in these plots (Figures 10-1 to 10-7 in reference 41), it can be stated conclusively, independant of pump suction void fractions, that the highest vibrations observed during draining were caused by pump head instabilities. The frequencies of the head pulsations measured during the high vibration periods are not mentioned in reference 41 but Fig. 9 (from that reference) shows the measured amplitudes. As seen in Figure 9, the pulsation amplitudes measured in the tests performed with initial temperatures above 170°C were less than 10% of the amplitudes recorded at temperatures below 140°C. The dramatic change in pulsation amplitude that appears to have occurred between temperatures of 140°C and 170°C suggests that the mechanism responsible for the head pulsations observed at temperatures above 170°C was fundamentally different from the mechanism that caused the pulsations at lower temperatures.

Note that correlating vibration to temperature tells only part of the story. The void compliance vapor/liquid density ratio, etc. are more closely related to suction pressure than temperature under voiding conditions so plots of vibration versus pressure would probably be more instructive than vibration versus temperature.

High levels of vibration were observed in test 194 while the test loop was being refilled with cold water. Since the loop temperature at the time of the vibrations was 100°C, and the vibrations occurred while the pump head was recovering (i.e., towards the end of the refill period), there does not seem to be any reason to suspect that the cause of the vibrations was any different from the cause of the vibrations observed during bleeding at temperatures below 140°C Although it is possible that the injection of cold water into the loop de-stabilized the flow somewhat, it is unlikely that the effect would have been very large once the pump head had started to recover because the refill rate (which was limited to a maximum value of 6 l/s) would have become a very small fraction of the main loop flow once head recovery had started. As seen in Fig. 10, which shows the loop flowrate measured during bleeding as a function of time for a test performed with a starting temperature of 100°C, the loop flowrate was at least 0.15 m^3/s (150 l/sec.) before the pump became completely steam blanked and flow stopped altogether. Since a flowrate of 150 l/s is 25 times the maximum refill rate of 6 l/s, and the cold water was injected into the Pickering test loop approximately 7.6 m (i.e. approximately 25 pipe diameters) upstream of the pump suction flange, the main loop flow and the refill flow would probably have been well mixed and in approximate thermal equilibrium by the time the flow entered the pump. Thus it is considered unlikely that the pump behaved differently during refill than it did during bleeding. It is well known that the injection of cool water into a hot loop can cause flow instabilities by itself (Tong, Ref. 43, for instance) but it is considered unlikely that such injection affected the pump's "forcing function" in either the Bruce or the Pickering pump tests for the reasons just mentioned. There is no strong evidence to suggest that the injection of cool water into either of those loops affected their dynamic hydraulic response by some mechanism other than by simply lowering the water temperature. (That is, by some mechanism which would have caused head pulsation amplitudes and frequencies to be different during bleeding than during filling at the same temperature.)

E.4 Air - Water Tests of Rothe et al. (Ref. 42)

Rothe et al. tested a 1/20th scale model of a nuclear reactor primary coolant pump with the performance characteristics given below in the air - water test loop shown in Fig. 11.

Rated flow	-	13.9 l/sec
Rated head	-	75 m
Pump speed	-	18000 rpm
Specific speed	-	4250 $\frac{\text{rpm US gpm}^{1/2}}{\text{ft}^{3/4}}$

The tests were performed to study pump surge in two-phase flow and to obtain experimental data for validating an analytical model of pump surge developed by Rothe et al. (Ref. 44) in earlier work. The analytical model developed is described in Section G.

The tests were conducted at a nominal loop pressure of 0.3 MPa, at pump speeds of 0, 3000, 6000 and 9000 rpm, which were 0%, 16.5%, 33% and 50% of the pump's design speed of 18000 rpm respectively. Tests were performed over a range of air and water flow rates at each pump speed, although the water flowrate was kept below 50% of the pump's design flow (13.9 l/s) because oscillatory flow could not be obtained at higher liquid flowrates.

Two kinds of oscillatory flow were identified:

(1) at low to moderate void fraction (less than 40%), flow surges in the loop piping; and,

(2) at higher void fractions, flow oscillations typical of slug flow

Real surge behaviour was most evident in the tests performed at void fractions below 20%. Other conclusions drawn from the tests were:

(1) Surge would not occur at void fractions below some critical value. The critical value was approximately 5% for the pump tested.

(2) The flow became more stable as the total flowrate through the pump increased.

(3) The flow became more stable as the pump speed decreased and, moreover, that oscillatory flow would not occur below some critical pump speed that depended on the void fraction of the flow At a void fracton of 10%, for instance, the critical pump speed was in the 3000 - 6000 rpm range (or 16.5% - 33% of design speed).

(4) The transition boundary between steady and surging flow was insensitive to the nominal pressure of the system. However, pressure pulsation amplitudes and frequencies during surge both increased as the nominal system pressure increased.

Since all of the tests performed by Rothe et al. were conducted at pressures below 0.5 MPa (and hence at extremely low vapour-to-liquid density ratios), the aforementioned conclusions (4, at least) would not necessarily hold for high-pressure steam-water circuits.

F. EFFECT OF SYSTEM PARAMETERS

F.1 Flowrate

The results of the Bruce pump tests showed that pipe vibration levels decreased as flowrate increased (because refill rates were relatively low, comparison to ECI cannot be conclusive). Lower flowrates were not run for the Pickering tests. Rothe et al. (Ref. 42) observed that flow stability increased as flowrate increases, showing qualitative agreement with the results obtained by Ontario Hydro in the Bruce pump tests. See Section G.2.

In two-phase flows of moderate void fraction, hydraulic damping increases with flow (due to increased flow resistance). This factor may contribute to increased stability at higher flows.

Since the reviewed literature generally shows a lack of data or quantitative discussion on intermittent or slug flow, these regimes are not discussed in any detail in this report.

F.2 Pressure/Temperature

The effects of average system pressure and temperature on pipe/flow oscillation will be considered together since, for steam-water systems, pressure and temperature are interdependent under equilibrium conditions. (Note, however that under voiding conditions, pressure is a better measure of system conditions.)

The results of the Bruce and Pickering pump tests as well as the results obtained by Rothe et al. show that the frequency of pump-induced pressure pulsations in a two-phase system increases as system pressure increases. The results of the Bruce and Pickering pump tests appear to be contradictory on the question of when high head pulsations occur, since high head pulsation amplitudes caused by pump head instabilities were observed (1) at temperatures **above** 170°C in the Bruce pump tests and (2) at temperatures **below** 140°C in the Pickering pump tests.* Oscillating head instabilities were observed at temperatures between 100°C and 140°C in the Bruce B pump tests. The head pulsations that occurred in the Bruce pump tests when the system temperature was approximately 180°C caused the most severe vibrations of that loop because the head pulsation frequency at that temperature was close to a fundamental frequency of the loop piping.

---

* This could be a result of design differences between the pumps, or different localized voiding conditions (due to differences in the loop geometries).

The pump surge test results obtained by Rothe et al. (Ref. 42) are in qualitative agreement with the Bruce pump test results in that Rothe et al. observed an increase in the amplitude of the head pulsations as the system pressure was raised. When one looks at Figures 5 and 9, however, it is apparent that, for a given loop geometry flow resistance and pump, the system pressure effectively governs whether pump head instabilities occur at all (i.e., there is no gradual reduction or increase in head pulsation amplitude as temperature and hence pressure is increased). This observation contradicts the fourth conclusion of the study of Rothe et al. ("The transition boundary between steady and surging flow is predicted to be insensitive to the nominal pressure of the system,...")

Schafer and Simon (Ref. 29) state that "at elevated temperatures pump performance is much less sensitive to cavitation than at low temperatures". Vibration increases with decreasing suction head at low temperature (50°C) but remains constant at high temperature (279°C). This can be attributed to a lower vapour-to-liquid density ratio at higher temperatures. The smaller vapour bubbles at high temperature do not inhibit flow as readily.

F.3 Void

The results of the Bruce and Pickering pump tests and Rothe et al. indicate that, if pump head instabilities occur at all, they will occur in the range of void fractions over which the pump head degrades rapidly. (Rapid head degradation never actually occurs at high mass flowrates - this is presumably why Rothe et al. did not observe oscillatory flow in their loop when the liquid flow rate was more than 50% of the design value). As observed in the SLOCAT 'B' pump test and as shown in Figure 5, other types of periodic and non-periodic instabilities (e.g. oscillating and intermittent flow instabilities) can occur at higher void fractions.

F.4 Voiding Rate

Reference 41 (Section 7.0) indicates that pump head instabilities did not occur in the Pickering NGS B pump tests when the void increase rate was higher than usual.* It appears, therefore, that voiding rate is a significant parameter, perhaps because the hydraulic conditions at a high voiding rate pass through critical regions (where instabilities are possible) too quickly for those instabilities to develop fully.

---

* The voiding rates are not quantified in the text of Reference 41, however they are shown graphically in Figure 11 of that report. The fastest rates correspond to the blowdown tests (no. 230 and 231).

G. ANALYTICAL AND EMPIRICAL MODELS

G.1 Analytical Models

Analysis of the problem could be broken into discrete models:

(1) pumpset structural/mechanical model;

(2) piping structural model;

(3) heat transport system thermal-hydraulic model;

(4) pump thermal-hydraulic model.

New analyses or simulation tools are not necessarily required for items (1) to (3). The existing tools are well developed, although they might require adaptation to the pump vibration problem.

Finite element tools work well for (1) and (2). Several piping, turbomachinery, rotor dynamic and shaft whirling analytical techniques are available (although they have not been investigated in detail here). Potential refinements would include modelling non-harmonic and non-linear phenomena, inelastic behavior, large deflections, solid-fluid interaction, etc.

Reactor thermal-hydraulic models (3) are also well established, although it is often not possible to model oscillatory phenomena without using prohibitively small space and time increments. Some codes show anomalies in certain conditons, such as pump head at critical void fractions.

With respect to (4), a number of empirical two-phase thermal-hydraulic models (eg. the Aerojet Nuclear Corporation model) exist, as attested by the Bibliography. Relatively few analytical models for predicting two-phase pump performance are available. Some of these models could possibly be improved, especially to simulate pump-induced flow oscillations.

G.2 Pump-Induced Flow Oscillations

Analytical models for predicting flow oscillations in a pump-piping system have been developed by Rothe et al. (Ref. 44) for a pump operating with pump suction voids in the "head instability" region, and by Rubin (Ref. 45) for a cavitating pump. Both of these models essentially determine the acoustic free-vibration characteristics (Ref. 46) of the pump-piping system from the pump steady-state "two-phase" or "cavitating" performance curves and thereby determine at which (if any) of the system's acoustic natural frequencies the flow would become unstable. In theory this procedure could be applied to any pump-piping system to predict conditions under which the system would become unstable, but an analysis which accounted for non-linear effects would have to be performed to predict actual pressure oscillation amplitudes, etc. (unless the flow perturbations were very small).

Only the analysis of Rothe et al. (Ref. 44) will be considered here since it is supported by experimental data.

The model of Rothe et al. applies to a loop in which virtually all of the flow resistance is concentrated at a single valve, as shown in Figure 12. A simple lumped-parameter model is developed by lumping all the compliances in the tank and all of the inertances in the pump discharge pipe. The pressure at the pump suction is assumed to remain constant at p_a.

An equation is described in Ref. 44 which could be used as a criterion for predicting flow stability resulting from small disturbances. That equation is:

$$\frac{dP}{dQ_p} < \frac{\rho L Q}{2AV} \qquad \text{(for stability)}$$

where:

P = pump discharge pressure

Q = total 2-phase volumetric flow rate

ρ = average density of 2-phase fluid*

---

* Note that ρ and V are average system values and may not reflect instantaneous, local values at the pump under two-phase conditions. The referenced paper does not describe exactly how these values are determined. Some verification of the values would be required to substantiate the results. (A similar drawback to the Ontario Hydro tests).

L = length of suction & discharge piping

Q_p = instantaneous 2-phase flow rate into pump

A = cross-sectional area of pump suction pipe

V = gas filled volume of piping*

Failure to satisfy this equation (i.e. the left hand side greater than the right hand side) leads to flow instability under certain conditions. This corresponds to the positive sloped curves (voids of 8%, 12% and 16%) in Fig. 13. Thus, in general, positive sloped curves indicate degredation of head.

For instance, as Q increases, the left hand side of the formula becomes increasingly negative while the right hand side becomes more positive. As a minimum, the curves shown in Figure 13 have to be positive in order for the flow to become unstable.

Note that such positive-sloped curves are normally characteristic of forward-swept impeller blades (centrifugal pump blades are backward swept) and also that Figure 13 is pump-specific. Since the above equation has to be violated for surge to occur, the difference between Bruce and Pickering test results could be explained by differences in the pumps characteristics. (Note the differences in specific speeds.) The pump characteristic curves of Figure 13, showing a positive slope with increasing void, were obtained using an unshrouded impeller (Ref. 47). Similar trends are seen in the data of Murakami and Minemura (Ref. 49).

Burchett has used the Rothe et al. model with some success (Ref. 35). He plotted surge frequency versus void for the Kipling tests, and found similar results. In Ref. 35 he suggests further testing to establish a correction factor for application of the Rothe et al. formula to the Ontario Hydro tests. (Subsequent testing or application of the equation was not apparent in the reviewed literature.)

Although Rothe et al. used a simple steady flow pump characteristic, actual surge behavior may depend on two-phase pump performance dynamics, hydraulic damping, the interaction of pump and piping, etc. More information on the variation of dP/dQ_p with pressure and in different pumps would aid in analyzing test results. Instability will be a function of pressure if the characteristic curves of Fig.13 vary with pressure.

Rothe et al. provide the following correlation for the head pulsation frequency: (W)

$$W = [PA/\rho LV]^{1/2}$$

Note that this equation predicts that W will increase as the system pressure (P) increases (in accordance with available experimental data).

G.3 Empirical Models

Due to the lack of information on other testing facilities, comments can only be made on Ontario Hydro's work.

The perceived limitations of Ontario Hydro's testing are listed below. Note that Ontario Hydro recognizes these limitations (see Appendix A). They are stated here for completeness and technical understanding, not as criticisms of the overall work.

(1) Similarity of piping geometry to the station does not extend beyond the first elbow . Extraneous fluid transients could be generated elsewhere in the loop. Examples of different geometries creating transients are the hairpins in the Bruce test loop. In fact, vibration occurred in the hairpins, limiting testing (see point (5) below).

(2) The loops cannot simulate interaction between multiple pumps in multiple loops, as in the stations.

(3) Structural supports of the motor and of adjacent piping do not match the station, possibly creating different frequency patterns in the lumped structure.

(4) The blowdown/refill capability of the Bruce loop is limited by the holding tank volume. Full ECI injection/slug flow was not simulated.

(5) Some tests in the Bruce loop were not completed due to resonant vibration of the piping. (Some consideration was given to supports and damping, but the physical and cost restraints were prohibitive.)

(6) Some low speed tests were not completed, in all likelihood due to "vapour locking" in the loop.

(7) Determination of thermal-hydraulic parameters (especially void fraction) at the pump is very difficult. Most parameters were estimated from loop average conditions. This factor made the analysis of flow instabilities other than surge (e.g. slug or intermittent flow) virtually impossible.

(8) Acquisition and reduction of data is a huge task which is none-the-less very important. Sampling rates are very important. A proper post-processor was not installed by Ontario Hydro until mid-way into the testing.

H. RECOMMENDED METHODOLOGY

Quantitative assessment of the forcing function acting on the pump-piping system due to void generation and collapse is very complex, but should be feasible. Several general comments can be made:

- Most of the relevant analyses unearthed in the literature search have some basis in empirical results; i.e. testing has played an important role in work to date.

- Surge phenomena require detailed analytic study. This is likely the major cause of head instability. Such a study is required before other instabilities can be analysed with any confidence.

- Extensive work has been carried out in two-phase thermal-hydraulics in pumps. Using steady-state pump performance characteristics obtained from tests, it is possible that surge behaviour could be predicted analytically in the manner described in Section G.

- Decoupling of fluid and mechanical forces is required to develop the initial analyses.

- If full LOCA tests were performed on a test loop identical to a real reactor, measurement of conditions at the pump would not be required; i.e. the pumpset performance could be verified. Building an identical test loop is prohibitively expensive, therefore thermal-hydraulic measurements at the pump are necessary as are subsequent analyses.

The proven approach to analyzing such complex behavior is to start with a fundamental model, refine and verify it, gradually increasing its complexity. Due to the detailed nature of the problem this work should be approached in a flexible manner; i.e. changes in mid-stream may be required. The steps required might be as follows:

(1) Verify the assumption of an uncoupled fluid/structural model. Although it seems to be a reasonable assumption, it has not been validated. (Note

that the analysis would be difficult. Consider, for example, the uncertainty in determining pressure amplitudes).

(2) Determine the steady-state two-phase performance characteristics of the pump over a range of inlet void fractions.

(3) Develop a hydro-acoustic model of the loop using the pump performance characteristics determined in (2); this could be a time domain analysis, based on the method of characteristics, or alternatively a frequency domain analysis based on periodic oscillations.

(4) Determine the forcing functions from results obtained with the hydro-acoustic model. A sensitivity study would be very useful to narrow the conditions (i.e. - assuming a full range of feasible hydraulic forcing functions in the pump casing and impeller to ascertain whether the resulting vibration responses are even of concern).

(5) Solve for the vibration of the loop piping with a finite-element model (e.g. NASTRAN or ANSYS).

(6) In parallel with steps (3), (4), and (5), develop a reduced scale test model using geometric similarity. Piping well upstream and downstream of the pump should be in an identical configuration to the real loop. The pump conditions should be measureable* to allow analysis of slug flow intermittent flow, etc. Return to steps (3) and (4) as required to refine the models for non-steady state conditions.

With respect to the test loop, a low pressure air-water loop would be least expensive. Air-water phenomena are also easier to model than steam-water conditions. The hydro-acoustic response could be developed by a code such as ACOUST (Reference 50).

A steam-water loop would be very expensive. The real phenomenon of condensation in the pump should be considered for a rigorous analysis. A time domain analysis by the method of characteristics could be employed, although development of an appropriate hydro-acoustic model (which included pipe branches, elbows, etc.) could be very difficult.

---

* e.g. a beryllium window for a gamma densitometer.

One option would be to build the test loop for high pressure (steam-water), but test initially with air-water to develop the analyses.

There is a lack of experimental data on intermittent and slug flow regimes. In Ontario Hydro's test program this was due to vapour locking conditions, insufficient NPSH and/or insufficient blowdown/refill capability. Development of a scaled test rig should account for these problems. For example, slug flow might be simulated by use of a heavy-duty quick-acting shutoff valve.

I. CONCLUSIONS AND RECOMMENDATIONS

I.1 Conclusions

I.1.1 Given the time and cost restraints of the contract, the literature search has shown that Ontario Hydro's work on the subject is the most thorough in the world.

I.1.2 Other sources may have done comparable work but the information is not readily available.

I.1.3 Several sources have semi-relevant data or analyses. Either vibration data has not been recorded methodically or the work objectives did not match the three major criteria of: two-phase flow, centrifugal pumps, and vibration. The effort required to apply this work to the problem would be quite substantial.

I.1.4 The mechanical/structural integrity of primary pumpsets is sound; they seem to be resilient to vibration, as shown by operating incidents (Section D.2) and the testing of Ontario Hydro. Seals and bearings may wear, but catastrophic failure has rarely (if ever) occurred. The thermal-hydraulic performance of pumps in two-phase flow is more in question than the mechanical integrity.

I.1.5 Results from Ontario Hydro's test program should be applied to actual plant analyses only after consideration of limitations, some of which are addressed in Section G.2. Especially applicable points are; (1) dissimilarity in piping geometry; and (2) difficulty in estimating thermal-hydraulic conditions locally at the pump.

I.1.6 Ontario Hydro's analyses are extensive and based on existing theory and actual test experience. However their program has some limitations. Some of the analytical limitations perceived include:

- the correlation of vibration and other parameters against temperature only;

- using **estimated** thermal-hydraulic conditions at the pump;

- disagreement of empirical analysis and SOPHT results;
- the assumption in plant analyses that piping is excited in the first mode only;
- lack of data for the Bruce tests at high voiding rates.

I.1.7 Pump head instability is perceived to be the most important region for inducing vibration. The emphasis of the reviewed literature attests to this point.

I.1.8 To improve understanding of the problem of two-phase flow induced vibration in pumpsets, a combined analytical and empirical approach is required, with initial emphasis on analysis.

I.1.9 The structural/mechanical aspects of the problem are not difficult to model. The difficulty lies in obtaining the hydraulic forcing function.

I.1.10 Testing can be deemed the only absolute proof of operation. However some factors must be considered:

- loop piping geometry should (ideally) be exact, as should pump characteristics, supports, etc; close simulation and scaling is of critical importance;
- loop dynamics cannot be recreated identically (i.e. reactor core, multiple pump interaction, steam generators, etc.);
- measurement of some thermal-hydraulic conditions at the pump is very difficult, if not impossible; and,
- data acquisition must be at very high sampling rates to retain all frequency responses, (which creates an increased effort in data reduction).

I.2 Recommendations

I.2.1 The literature review should be extended. Due to cost and scheduling restraints, many contacts could not be pursued, some documents have not yet been received, and some of the retained literature has not been thoroughly analyzed. The sources addressed in Section D should be given priority.

I.2.2 Questions remain on certain Ontario Hydro's tests (e.g. pressure rather than temperature data; low speed testing*). In these cases the raw data and initial analyses should be reviewed. Note that the volume of data is extensive. The cost-benefit of locating and reducing the appropriate data is highly dependant on the user's organization of this data, i.e. it should only be pursued if the data is readily available. Further testing is not seen as necessary.

I.2.3 A detailed analytical program should be initiated to quantify the fluid forcing function. A fundamental approach is recommended, assuming decoupling from structural analyses. (Coupling assumptions should be investigated as well).

I.2.4 At a later date, the fluid forces should be integrated into the other models (thermal-hydraulic and structural). These other models should be refinements of existing methods. Verification on a scaled test facility accurately modelling the desired geometry should be initiated in parallel with these theoretical analyses. (The proposed test facility is discussed in Section H.)

I.3 Limitations

Ontario Hydro's raw data has not been reviewed. Also, detailed analyses, other than those of Ontario Hydro, have not been obtained. Therefore most of the discussion and conclusions in this report are based on an overview of one institution's work.

---

* Pressure/temperature measurements for many tests, including Pickering NGS B test numbers 196-199, 201, 202, 206, 209, 237. Low speed tests including Bruce NGS A test numbers 8 and 81.

Due to the complex nature of the problem, proposed future work should be approached in a flexible manner; i.e. changes in approach in mid-stream may be required.

REFERENCES

(1) Bastien, P. & Kumar, A.N. Bruce 'B' Heat Transport Pump Performance Report, AECL, File 29-33122-200-00, 30/05/83.

(2) Black, H.F. Calculation of Forces Whirling and Stability of Centrifugal Pump Rotor Systems, ASME Paper 73-DET-131.

(3) Ontario Hydro, Design & Development Division, Personal Discussions, February 13, 14, 1984, (see Visit Report, Appendix A).

(4) Allaire, P., University of Virginia; Vibco Research Inc., Charlottesville, West Virginia, Personal Communications, March 1, 1984.

(5) Burchett, P. Review of Bruce A Commissioning Event on Unit 2 (Accident Blowdown) August 16, 1975 and Comparison with PHT Pump Test Data, Ontario Hydro File 907N-33120-2P.

(6) Opening of a Heat Transport System Relief Valve While Operating at Full Power Pickering N.G.S.,Ontario Hydro Significant Event Report 78-61, 16/05/78.

(7) Failure of Heat Transport Circulating Pump 3311-P6 Douglas Point G.S., Ontario Hydro Significant Event Report 75-20, 9/09/75.

(8) Analysis of the Three Mile Island - Unit 2 Accident, Technical Summary, NSAC - 80-1, Nuclear Safety Analysis Centre, EPRI, Palo Alto, California, Revised March 1980.

(9) Fagone, S., Letter from S. Ragone, Vepco, to A. Giambusso, AEC Docket # 50180-199.

(10) Incident Reporting System, CAORSO BWR, NEA/IRS - 109, 1981.

(11) Incident Report - 19, Dampierre, Nuclear Energy Agency, Feb 1980.

(12) Incident Reporting System, Reactor Feed Pump Trip, OECD Nuclear Energy Agency.

(13) Forrest, C.F., Westinghouse, Hamilton, Personal Communications, January 26 & February 24, 1984.

(14) Krishnan, Dr., AECL, WRNE, Pinawa, Personal Communications, January 26 & March 2, 1984.

(15) Wilson, Chan, et al. Analytical Models and Experimental Studies of Centrifugal Pump Performance in Two-Phase Flow, EPRI NP-677 Electric Power and Research Institute, Palo Alto, California. May 1979

(16) Kreps, D.A., Combustion Engineering, Windsor Connecticut, Personal Commnications, February 24, 1984.

(17) Kreps, D.A. & Kennedy W G. C-E/EPRI Tow-Phase Primary Pump Performance Program, Proc. of Topical Meeting on Thermal Reactor Safety, Sun Valley, Idaho, July 31 - Aug 4, 1977.

(18) Lee, L. and Chandra, S. Pump-Induced Fluctuating Pressure in a Reactor Coolant Pipe, Combustion Engineering Inc, Windsor, Conn., Interational Journal of Pres. Ves & Piping, Applied Science Publishers Ltd, England, November 1979.

(19) Kim, J., Electric Power Research Institute (EPRI), Palo Alto, California, Personal Communications, January 25, 1984.

(20) Schneck, R., University of Virginia, Charlottesville, West Virginia, Personal Communications, March 2, 1984.

(21) Marr, A. M-C., Phelps, R.L. & Katz, B. Loss of Component Cooling Water Capability of a PWE Reactor Coolant Pump, ASME Paper.

(22) Corniglion, J. & Albano, G. Theoretical and Experimental Study of Flow Through Centrifugal Pumps, Proc. Joint Symp. on Design and Operation of Fluid Machinery, Colorado State Univ., June 12-14, 1978.

(23) Grison, P. & Lauro, J.F. Experimental and Theoretical Investigations about Two-Phase Critical Flow Through a Pump, Polyphase Flow in Turbo-Machinery ASME, San Francisco, Dec 1978.

(24) Grison, P. & Lauro, J.F. Two-Phase Performance Characteristics of a PWR Primary Pump Under LOCA Conditions, Heat and Fluid Flow in Water Reactor Safety, I-Mech-E Conference Publications, 1977-8.

(25) Department Transferts Thermiques et Aerodynamique, L'Installation Diphasique a Rafales (Indira), Electricite de France, 8/5/79.

(26) Grison, P. Letter from P. Grison, to P. Burchett, Design & Development Division, Ont. Hydro, 28/3/80.

(27) Gibert, R.J. & Axisa, F. Flow Induced Vibrations of Piping, Transactions of the 4th International Conference on Structural Mechanics in Reactor Technology. Vol. F, San Francisco, California, 15-19 August, 1977.

(28) Gibert, R.J. & Axisa, F. Flow Induced Vibrations of Piping, Transactions of the 4th International Conference on Structural Mechanics in Reactor Technology, Vol.F., San Francisco, California, 15-19 August, 1978.

(29) Schafer, C. & Simon, C. Experimental Investigation of the Cavitation Behavior of Boiling Water Reactor Coolant Pumps, Kraftwerk Union, Germany.

(30) Schafer, H.R., Kraftwerk Union A.G., Karlstein Labs, Federal Republic of Germany, Telex to B.M Guthrie, dated February 28, 1984, (see Appendix E).

(31) Piplies, L., LOBI Project, ISPRA Establishment, Italy, Telex to B.M. Guthrie, dated February 24, 1984, (see Appendix E).

(32) Marenco, G., Worthington Desio, Italy, Telex to B M. Guthrie, dated February 24, 1984, (see Appendix E).

(33) Kirejczyk, J. Cavitation Signs in Rotodynamic Pump, Second International Conference on Cavitation, I Mech E Conference Publications 1983-4.

(34) Sebestven, Gy., Rizk, M. & Szabo, A. Some Properties from Cavitation - Vibration Tests on Centrifugal Pumps,Proc. Joint Symposium on Design & Operation of Fluid Machinery, Vol. II, Colorado State University, June 10-14, 1978.

(35) Burchett, P.R. and Sauve, R.G. Report on Pipe Vibration During Simulated Loss of Coolant Test on Bruce Pump at Kipling Ave, Design and Development Division Report 82194.

(36) Koehler, H.P. Small Break Loss-of-Coolant Accident Test Bruce NGS 'A' PHT Pumpset Rotor, Hydraulic and Pipe Vibration, Ontario Hydro Research Division Report 82-239-H, 16/07/82.

(37) Burchett, P.R., Brown, N.G. and Lyons, R.G. Ontario Hydro Research Centre Pump Test Complex Heat Transport Pump Test Loop Facility, Design and Development Division Report 80262.

(38) Lin, T.C Frequency Response Analysis of Bruce GS 'B' PHT Pump Discharge Piping System Under Loss of Coolant Accident Conditions, Design and Development Division Report 83260.

(39) Koehler, H.P. Simulated Loss-of-Coolant Accident Tests: Bruce NGS 'B' Pumpset: Rotor, Hydraulic and Pipe Vibration, Ontario Hydro Research Division Report B83-64-H, 14/10/83.

(40) Boyd, J.M. Simulated Loss-of-Coolant Accident Tests Pickering NGS 'B' PHT Pumpset Rotor, Hydraulic and Pipe Vibration, Ontario Hydro Research Division Report 83-483-H, 12/12/83.

(41) Lockett, J.C , Toong, T. and Burchett, P.R. Pickering NGS 'B' Assessment of Primary Heat Transport Pump Loss of Coolant Accident Tests, Design and Development Division Report 83328.

(42) Rothe, P.H., Runstadler, P.W. and Doan, F.K. Pump Surge due to Two-Phase Flow, New York, A.S.M.E., Presented at the Winter Annual Meeting of the A.S.M.E. San Francisco, California, Dec 10-15, 1978.

(43) Tong, L.S. and Wieshan, J. Thermal Analysis of Pressurized Water Reactors, Second Edition, Illinois, American Nuclear Society, 1979.

(44) Rothe, P.H. & Runstadler, P.W. First - Order Pump Surge Behavior, ASME Paper 77-WA/FE-12.

(45) Rubin, S. Longitudinal Instability of Liquid Rockets Due to Propulsion Feedback (POGO), Journal of Spacecraft and Rockets, 3, No. 8, 1966.

(46) Wylie, E.B. and Streeter, V.L. Fluid Transients, New York, McGraw - Hill, Inc. 1978.

(47) Patel, B.R. & Runstadler, P.W. Investigations into the Two-Phase Flow Behavior of Centrifugal Pumps, Polyphase Flow in Turbomachinery, Winter Annual Meeting of the ASME, San Francisco,California, Dec 10-15, 1978.

(48) Three Mile Island Task Group, Guthrie, B.M. (Editor), Nuclear Studies and Safety Department, Ontario Hydro, A Review of Recommendations Arising from the Three Mile Island Accident, August 1980.

(49) Murakami, M. and Minemura, K., Effects of Entrained Air on the Performance of a Centrifugal Pump, Bulletin of JSME, Vol. 17, No. 110, August 1974.

(50) Marshall, A. ACOUST - A Program to Calculate One-Dimensional Wave Behaviour in Pump/Pipe Circuits of Arbitrary Connection Geometry, CEGB Report, 1978/10.

ACKNOWLEDGEMENTS

The assistance of A. Omar, H. Ragheb and G. Ishack of the AECB is greatfully acknowledged. R.G. Lyons, P.R. Burchett, D.K. Vijay and R. Hartlan of Ontario Hydro also provided valuable help in this work.

Mode No	Frequency (Hz)		Δd Max. Allowable Displacement at the North Hairpin Measuring Station MC, for 10^5 Cycles (mm peak-to-peak)	Δd Measured Displacement at Station MC (mm peak-to-peak)
	Without Loop Top Interconnecting Pipes	With Loop Top Interconnecting Pipes		
1	1.44	1.44	92	?
2	1.55	1.55	85	70*
3	1.88	1.91	21	?
4	1.92	1.98	14	?
5	2.71	2.71		
6	2.82	2.92		
7	4.34	4.34		

* The value of 70 mm peak-to-peak was used as a test limit and when this value was reached test conditions were revised so the limit would not be exceeded.

10^5 cycles at 1.55 Hz represents 17.9 h.

TABLE 1 (Ref. 36)

CALCULATED NATURAL FREQUENCIES, ALLOWABLE DEFLECTIONS FOR THE PUMP TEST LOOP MODEL ASSUMING NO WATER IN THE SYSTEM, AND OBSERVED DEFLECTIONS
BRUCE NGS A PUMP TESTS

Test #	System Initial Temperature (^{o}C)	Indicated Flow (m^3/s)	Max. Vibr. Amplitude (mm)
73	212	1.33	60
84	209	2.52	60
72	207	5.05	20
82	202	5.04	20
87	195	3.30	50
83	191	4.37	30
71	183	0.88	70
85	182	3.30	35

(a) All tests performed at full speed.

(b) Maximum vibration always occurred when cold water

(c) Vibration measurement on "main loop piping"

(d) Table transcribed from Reference 35.

TABLE 2 MAXIMUM VIBRATION AMPLITUDES MEASURED AT DIFFERENT FLOWRATES IN THE BRUCE NGS A PUMP TESTS

TEST NO	DATE	TYPE	TEST NO	DATE	TYPE
189	83 03 24	Commissioning 55°C	216	83 04 22	Blowdown
190	83 03 25	Commissioning	217	83 04 22	Blowdown
191	83 03 29	Commissioning 250°C	218	83 04 22	Rundown
192	83 03 30	Commissioning	219	83 04 22	Rundown 250°C
193	83 04 05	Commissioning	220	83 04 22	Rundown 250°C
194	83 04 06	Void Matrix 250°C	221	83 04 25	Transient 250°C
195	83 04 07	Void Matrix 220°C	222	83 04 25	Transient 250°C
196	83 04 11	Void Matrix 200°C	223	83 04 25	Transient 250°C
197	83 04 12	Void Matrix 250°C	224*	83 04 26	Transient 250°C
198	83 04 12	Void Matrix 220°C	225	83 04 26	Transient 250°C
199	83 04 13	Void Matrix 180°C	226	83 04 27	Transient 250°C
200	83 04 15	Calibration	227	83 04 27	Transient
201	83 04 18	Void Matrix 140°C	228	83 04 27	Aborted
202	83 04 18	Void Matrix 100°C	229	83 04 27	Transient 250°C
203	83 04 18	Blowdown 250°C	230	83 04 27	Transient 250°C
204	83 04 19	Blowdown 250°C	231	83 04 27	Transient 250°C
205	83 04 19	Blowdown 250°C	232*	83 04 28	Void Matrix 100°C
206	83 04 19	Void Matrix 100°C	233*	83 04 28	Void Matrix 250°C
207	83 04 19	Transient	234	83 04 29	Void Matrix
208	83 04 20	Blowdown 250°C	235	83 04 29	Void Matrix
209	83 04 21	Transient	236*	83 05 02	Blowdown 250°C
210	83 04 21	Transient	237	83 05 02	Void Matrix 56°C
211	83 04 21	Transient	238	83 05 03	Blowdown 250°C
212	83 04 21	Transient	239	83 05 05	Var Flow 120°C
214	83 04 21	Rundown	240	83 05 05	Var Flow 120°C
215	83 04 22	Blowdown	241	83 05 05	Var Flow 245°C

The pump ran a total of 77 hours.

*Movies made.

TABLE 3 (Ref. 40)

SUMMARY OF PICKERING NGS B PHT PUMPSET

SIMULATED LOSS OF COOLANT ACCIDENT TESTS

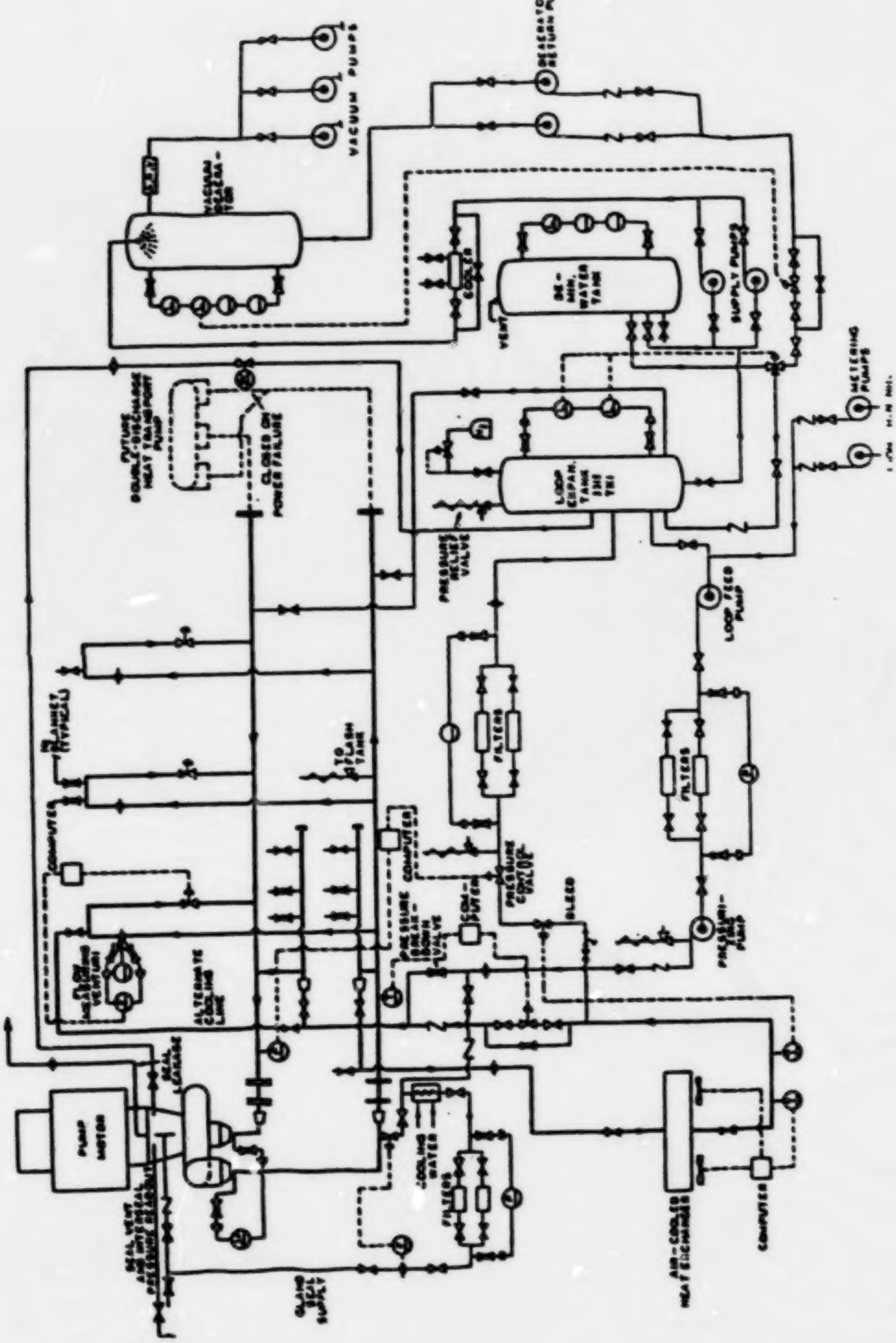
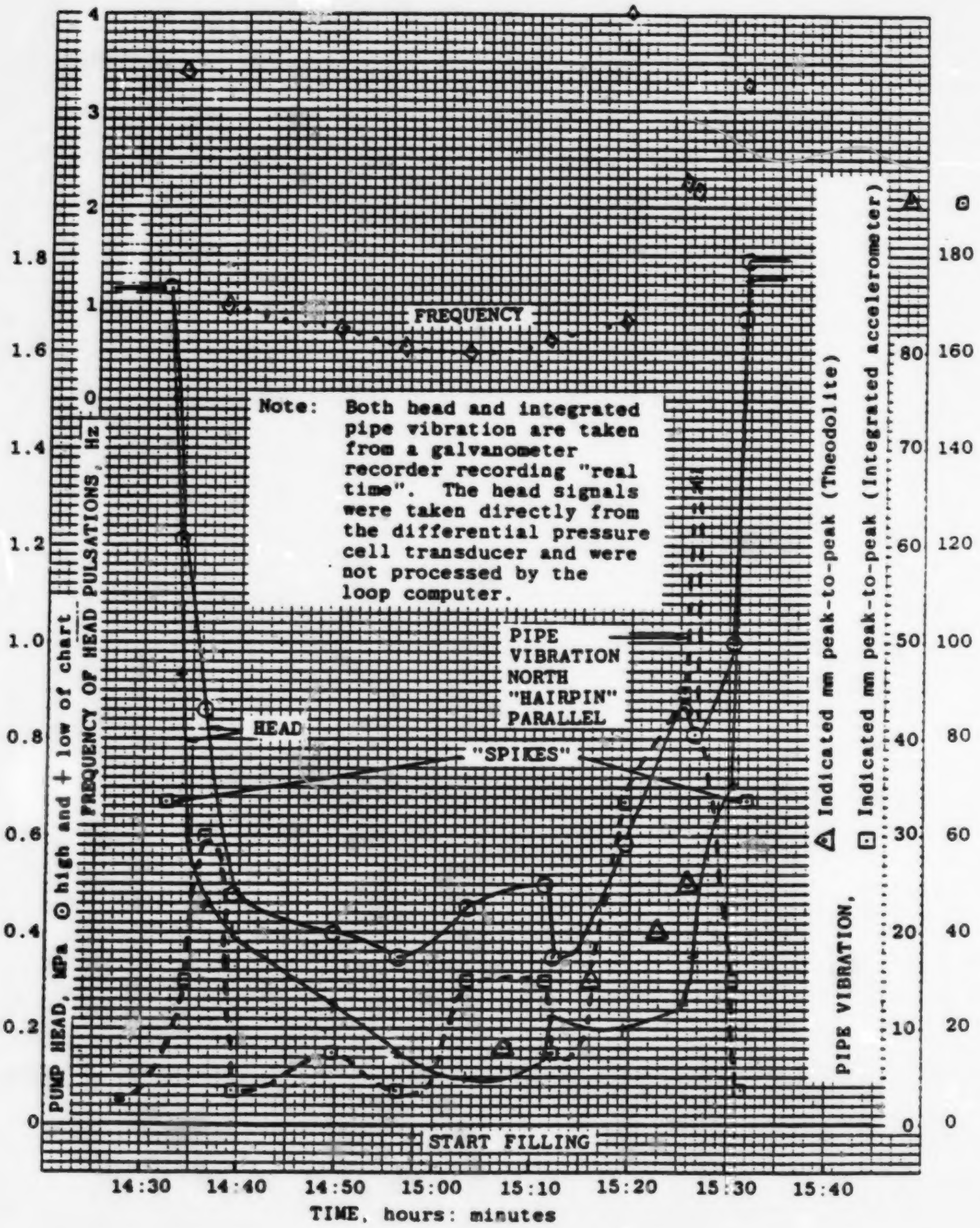

VACUUM PUMPS
DEAERATOR RETURN PUMP
VACUUM DEAERATOR
COOLER
DE-MIN. WATER TANK
VENT
SUPPLY PUMPS
METERING PUMPS
FUTURE DOUBLE-DISCHARGE HEAT TRANSPORT PUMP
CLOSED ON POWER FAILURE
LOOP EXPAN. TANK
PRESSURE RELIEF VALVE
LOOP FEED PUMP
FLANGES (TYPICAL)
TO SPLASH TANK
FILTERS
COMPUTER
PRESSURE CONTROL VALVE
BLEED
PRESSURE BREAK-DOWN VALVE
PRESSURIZING PUMP
FLOW MEASURING VENTURI
ALTERNATE COOLING LINE
SEAL LEAKAGE
PUMP MOTOR
COOLING WATER
SEAL VENT AND INTERNAL PRESSURE BREAKDOWN
GLAND SEAL SUPPLY
AIR-COOLED HEAT EXCHANGER

FIGURE 1 (Ref. 37)

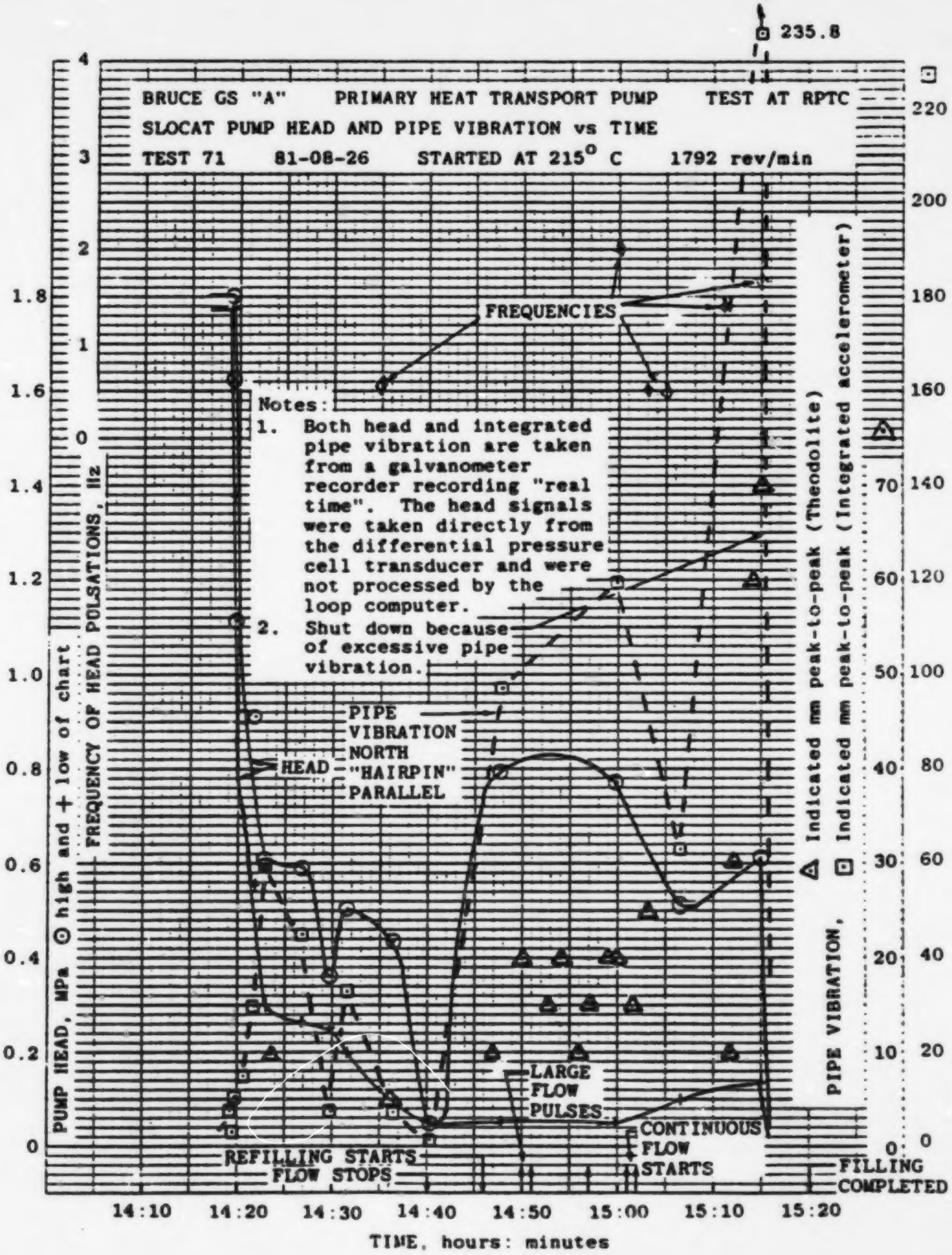
PHT PUMP TEST LOOP - FLOW DIAGRAM

SLOCAT PUMP HEAD AND PIPE VIBRATION vs TIME   BRUCE G. A
TEST 69   81-08-20   STARTED AT 235° C   1792 rev/min



Note: Both head and integrated pipe vibration are taken from a galvanometer recorder recording "real time". The head signals were taken directly from the differential pressure cell transducer and were not processed by the loop computer.

FIGURE 2 (Ref. 36)



BRUCE GS "A" PRIMARY HEAT TRANSPORT PUMP TEST AT RPTC
SLOCAT PUMP HEAD AND PIPE VIBRATION vs TIME
TEST 71 81-08-26 STARTED AT 215° C 1792 rev/min

Notes:
1. Both head and integrated pipe vibration are taken from a galvanometer recorder recording "real time". The head signals were taken directly from the differential pressure cell transducer and were not processed by the loop computer.
2. Shut down because of excessive pipe vibration.

235.8
FREQUENCIES
HEAD
PIPE VIBRATION NORTH "HAIRPIN" PARALLEL
LARGE FLOW PULSES
CONTINUOUS FLOW STARTS
REFILLING STARTS
FLOW STOPS
FILLING COMPLETED
PUMP HEAD, MPa ⊙ high and + low of chart
FREQUENCY OF HEAD PULSATIONS, Hz
PIPE VIBRATION, △ Indicated mm peak-to-peak (Theodolite)
⊡ Indicated mm peak-to-peak (Integrated accelerometer)
TIME, hours: minutes
0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
0
1
2
3
4
0
10
20
30
40
50
60
70
0
20
40
60
80
100
120
140
160
180
200
220
14:10
14:20
14:30
14:40
14:50
15:00
15:10
15:20

FIGURE 3 (Ref. 36)

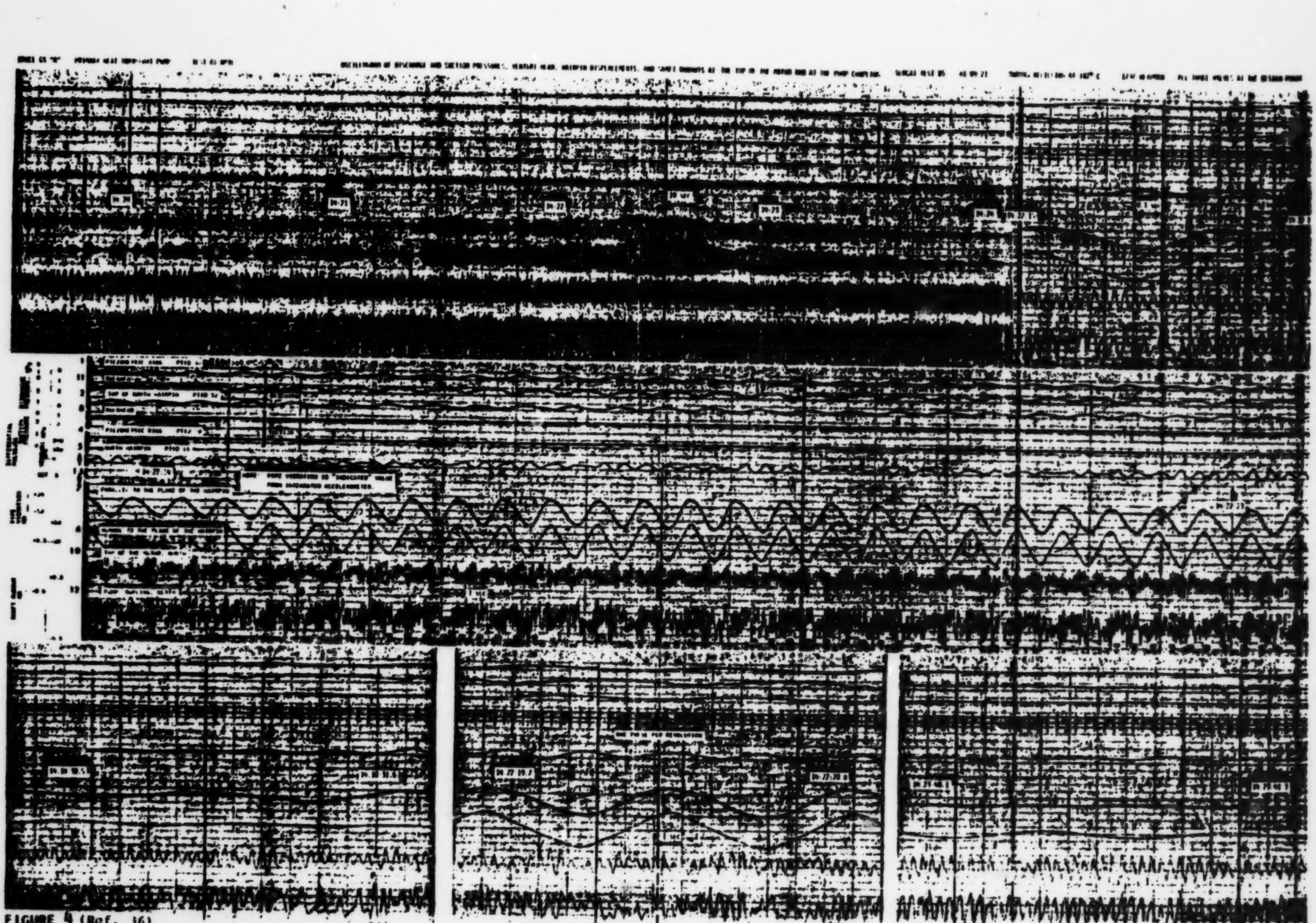
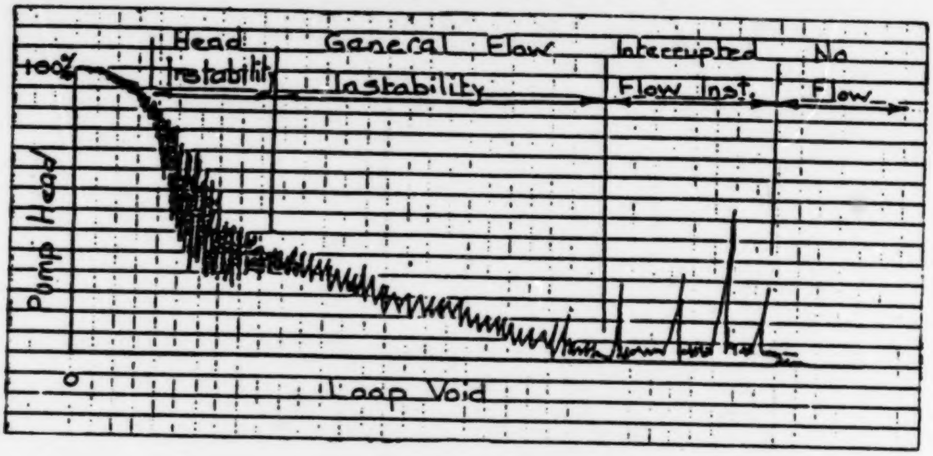
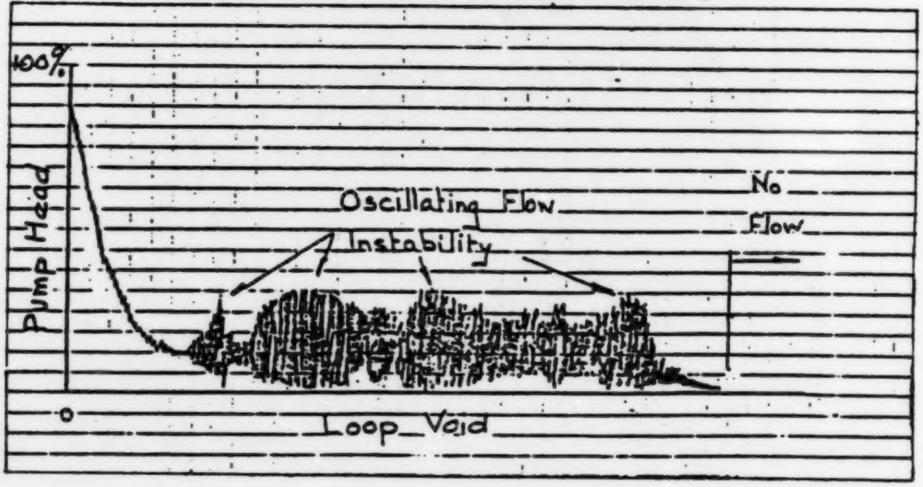


FIGURE 4 (Ref. 36)



Temperature Range 265°C to 170°C



Temperature Range 140°C to 100°C

FIGURE 5

PICTORIAL PRESENTATION - PUMP HEAD VARIATION V'S VOID - SHOWING AREAS OF INSTABILITY - BRUCE NGS 'B' (Ref. 38)

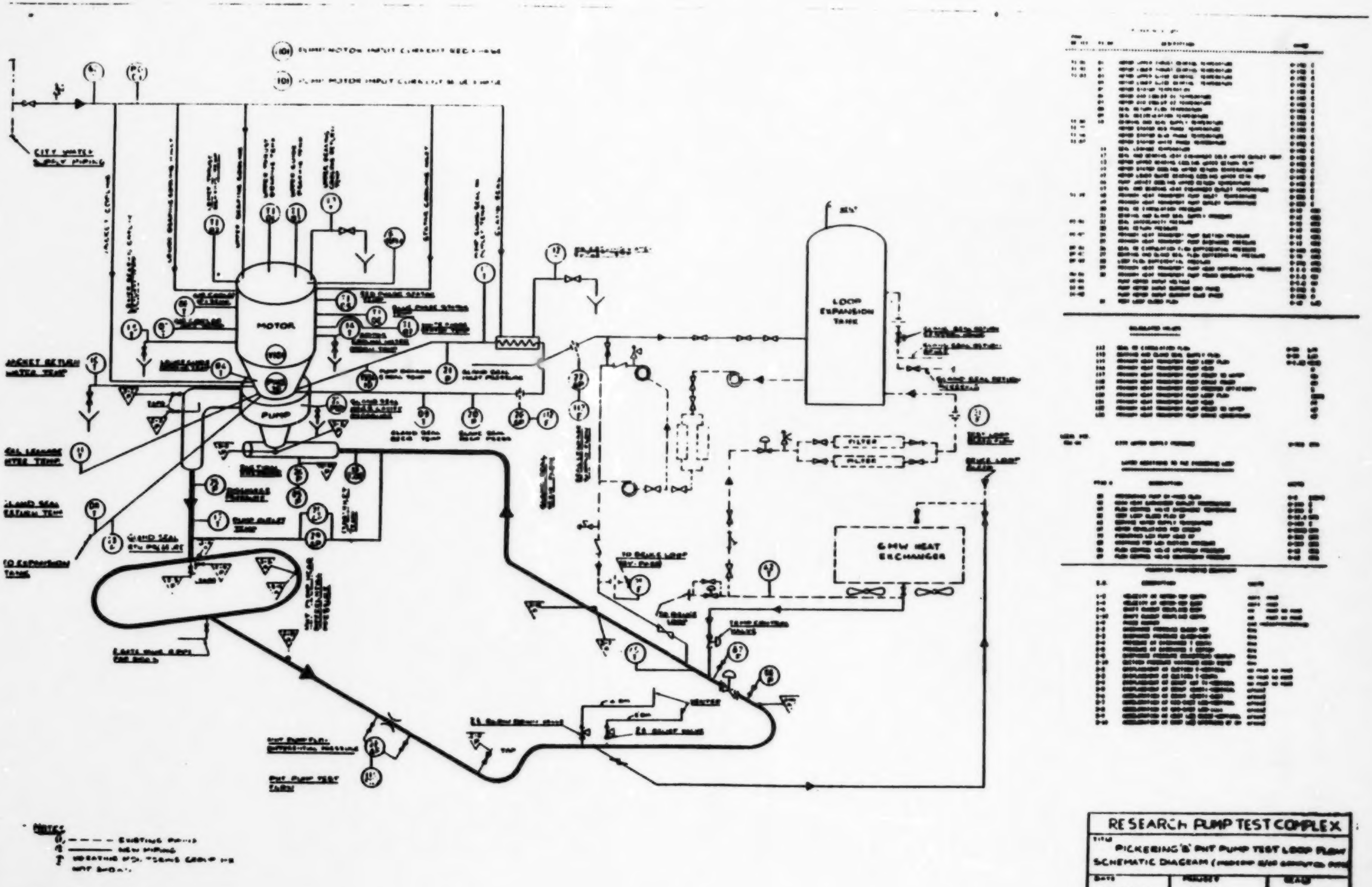

CITY WATER SUPPLY PIPING
JACKET COOLING
MOTOR
PUMP
JACKET RETURN WATER TEMP
GLAND SEAL EXHAUST TEMP
TO EXPANSION TANK
GLAND SEAL
LOOP EXPANSION TANK
FILTER
FILTER
G.M.W. HEAT EXCHANGER
NOTES
EXISTING PIPING
NEW PIPING

FIGURE 6 (Ref. 41)

RESEARCH PUMP TEST COMPLEX

TITLE
PICKERING 'B' PHT PUMP TEST LOOP FLOW
SCHEMATIC DIAGRAM

DATE	PROJECT	SCALE
MAY 4 81	22-131	NTS

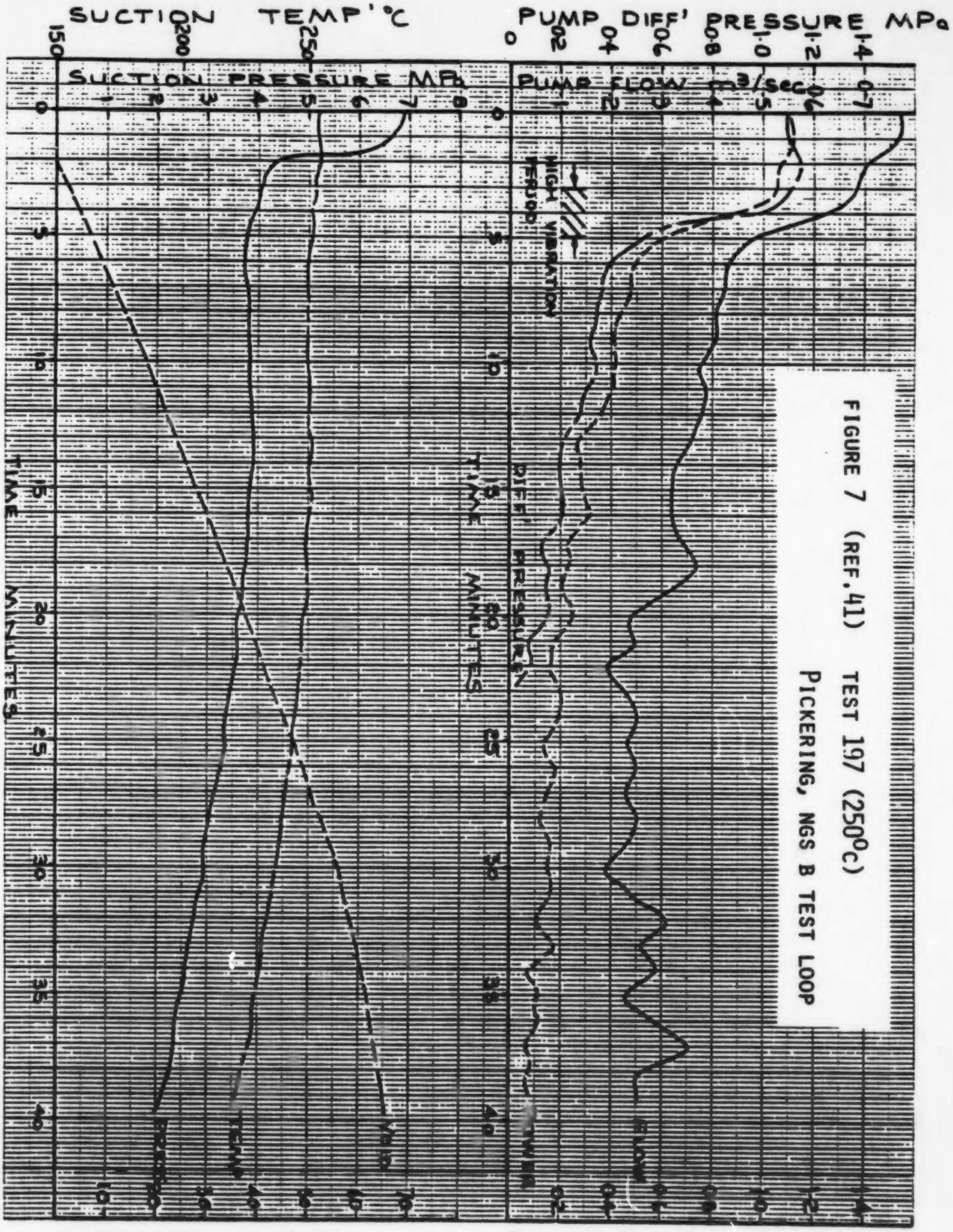


FIGURE 7 (REF.41) TEST 197 (250°C)
PICKERING, NGS B TEST LOOP

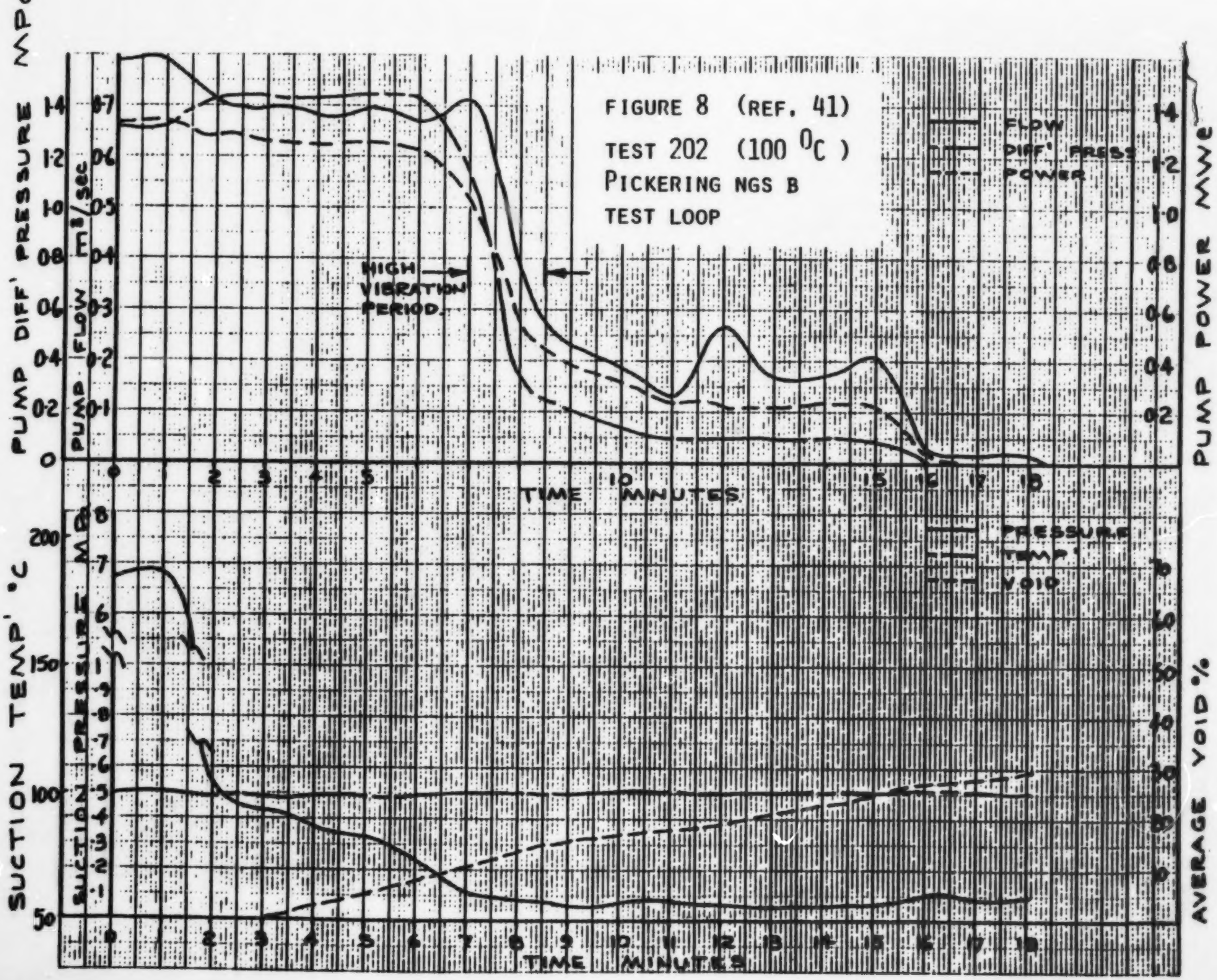


FIGURE 8 (REF. 41)
TEST 202 (100 ^{0}C)
PICKERING NGS B
TEST LOOP

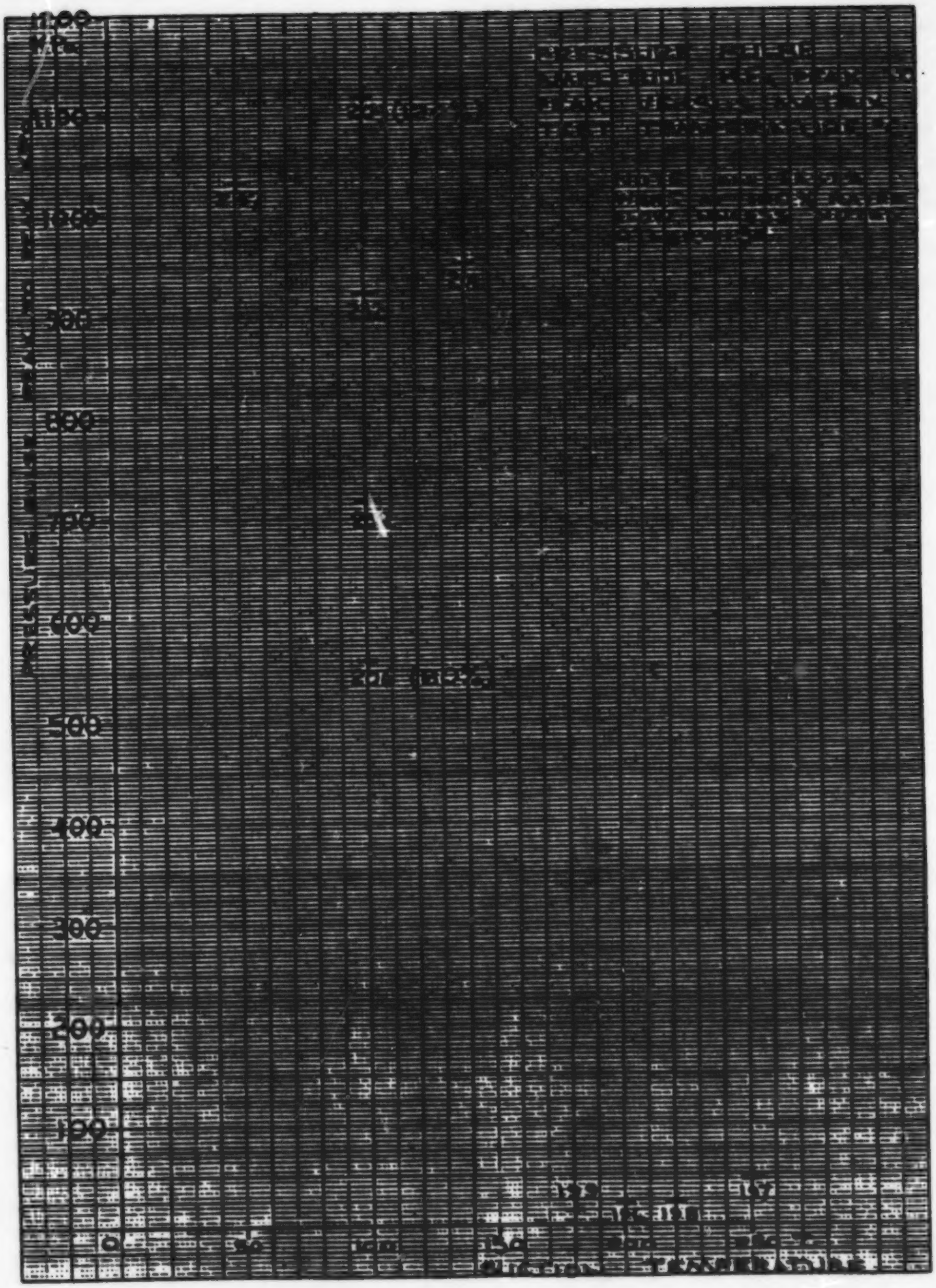


FIGURE 9 (Ref. 41) PRESSURE PULSE AMPLITUDE VERSUS MATRIX TEST TEMPERATURE

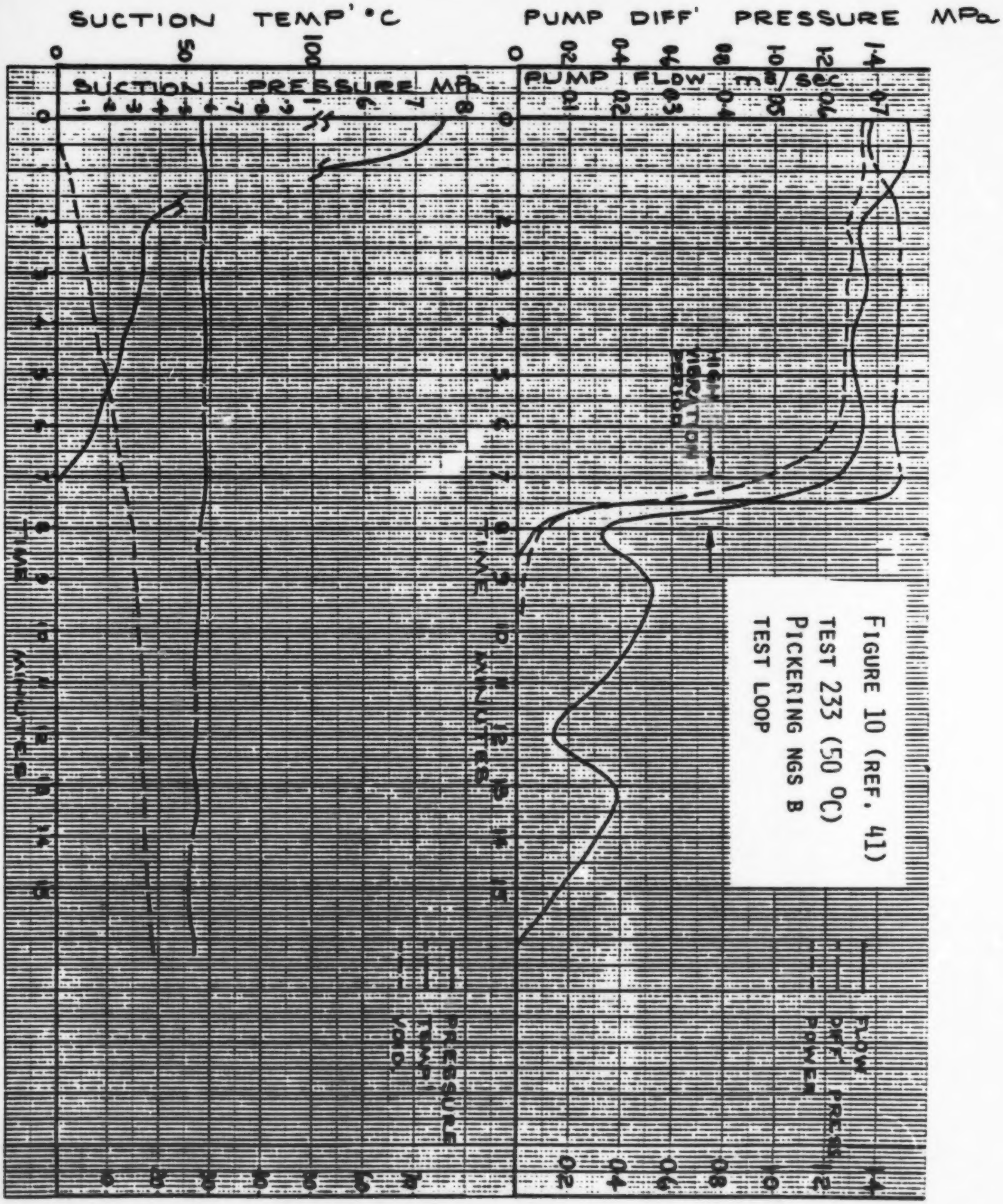


FIGURE 10 (REF. 41)
TEST 233 (50 °C)
PICKERING NGS B
TEST LOOP

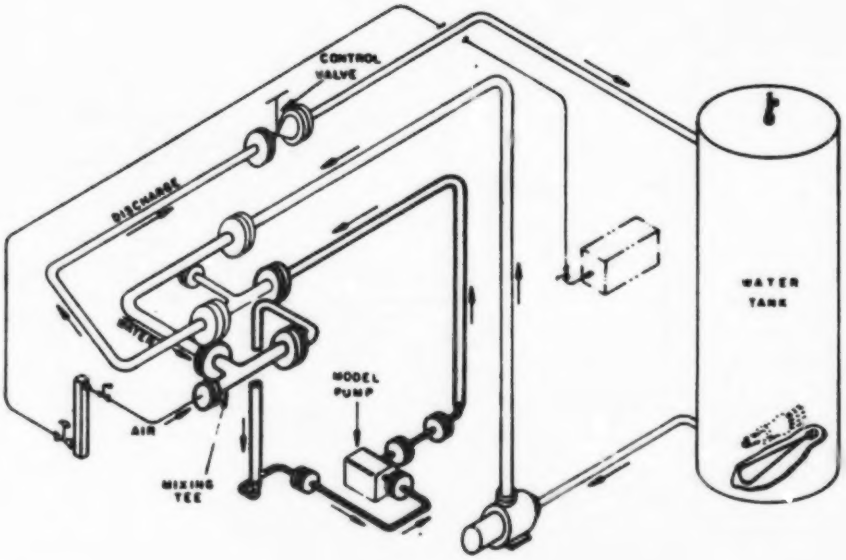

CONTROL VALVE
DISCHARGE
WATER TANK
MODEL PUMP
AIR
MIXING TEE

FIGURE 11 SCHEMATIC OF TEST FACILITY
ROTHE, ET AL (Ref. 42)

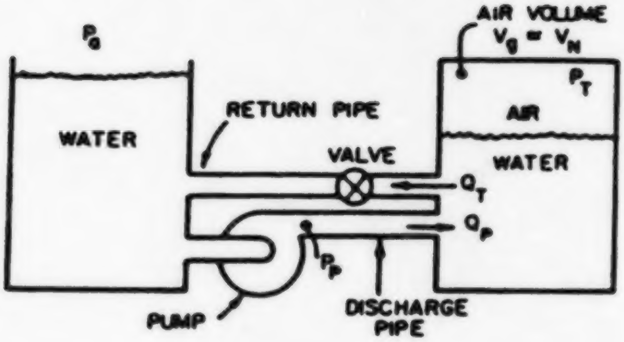
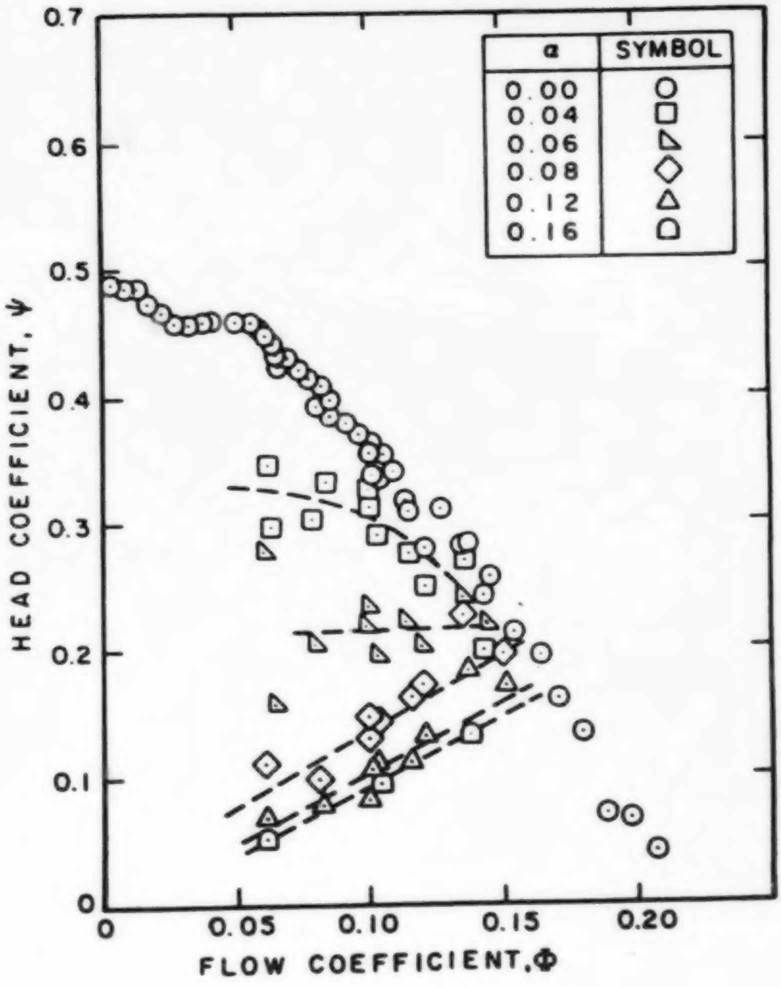

P_a
AIR VOLUME
$V_g = V_N$
P_T
RETURN PIPE
AIR
WATER
VALVE
WATER
Q_T
Q_P
P_P
PUMP
DISCHARGE
PIPE

FIGURE 12 IDEALIZED PUMP SURGE SYSTEM
ROTHE, ET AL (Ref. 44)

α = estimated average void fraction



$$\psi = \frac{\Delta P_b}{\rho_f U_2^2}$$

$$\Phi = \frac{Q}{2\pi r_2 b_2 U_2}$$

FIGURE 13 CROSSPLOT OF PUMP PERFORMANCE DATA FROM AIR-WATER TESTS (Ref. 47)

APPENDIX A

List of Activities, Searches, Contacts

1. **Canadian Institute for Scientific and Technical Information CISTI (NRC Library)**

   - full searches on three data bases;
   - NTIS - National Technical Information Services (U.S. Government Reports);
   - INIS - International Nuclear Information Services;
   - Engineering Index;
   - browsing through relevant sections of the library.

2. **AECB LIBRARY**

   - borrowed copies of relevant sections of Safety Reports, Design Manuals and miscellaneous reports, submitted by the utilities (Pickering 'B', Bruce 'B', Darlington, Pt. Lepreau and Gentilly-2 stations);
   - cursory review of Ontario Hydro's Commissioning Reports and Quarterly Reports;
   - search of card indices;
   - review of NUREG, NTIS, AECL reports and abstracts;
   - browsing through relevant sections of library.

3. **GEORGE ISHACK, AECB**

   - review of Incident Reporting System (IRS) and Ontario Hydro SER's, 1982/83.

4. **HELMI RAGHEB, AECB**

- reports on seal failures.

5. **PERIODICALS 1978-1984**

- Nuclear Safety;
- Nuclear Engineering International;
- Nuclear Engineering & Design;

6. **ONTARIO HYDRO VISIT**

- see visit report, Appendix C

7. **TELEPHONE CONVERSATIONS**

7.1 **Electric Power Research Institute, California,**

- Jong Kim, Project Manager, Nuclear Division

- Documents Centre - sending reports

7.2 **U.S. Department of Energy, Office of Converter Reactors,**

- Dennis Harrison

7.3 **CGE/Dominion Engineering, Research, Lachine,**

- Nigel Murray

7.4 **Westinghouse Canada, Hamilton,**

- Fraser Forrest

7.5 **Westinghouse Water Reactor Division, Chestwick, Pennsylvania,**

- Harvie Allen, RCP Engineering

7.6 Byron Jackson, Toronto,

- Mr. Padgett, Engineering Dept

7.7 Bingham Williamette, Virginia,

- Dwayne Morris

7.8 Serdula Systems, Consultants, Deep River,

- Ken Serdula

7.9 AECL - WRNE, Thermal-Hydraulics Research Branch,

- Dr. Krishnan

7.10 AECL - CRNL,

- Bert Carlucci
- Phil Ross-Ross
- Ray Metcalfe
- De Groenewald

7.11 AECL - EC,

- Arthur Earl
- Ralph Hart

7.12 ASEA, Montreal,

- Henry Sarberg, Industrial/Motors

7.13 Insurance Companies,

- Royal Ottawa
- Boiler Inspection Services
- Insurance Bureau of Canada

7.14 **Creare Ltd., New Hampshire,**

- Walt Smith

7.15 **Combustion Engineering, Windsor, Conneticut,**

- Denny Kreps

7.16 **Vibco Research Inc./University of Virginia Charlottesville, West Virginia,**

- Paul Allaire
- Ralph Schneck

7.17 **McMaster University, Dept of Engineering Physics, Hamilton,**

- Bill Garland

7.18 **Vibration Engineering Ltd., Oakville, Ontario,**

- Bob Wilson

7.19 **Mechanical Technology Inc, Latham, New York,**

- Chester Lee

7.20 **Science Applications Inc, Rockville, Maryland,**

- Mike Higgins

7.21 **University of Ottawa, Ottawa,**

- Danny Gorman

8.0 TELEXES

8.1 L. Piplies
Ispra Establishment
Div. of Ht. Transfer and Fluid Dynamics
I - 21020 Ispra (Varese), Italy

8.2 S.G. Marenco
Worthington Nord SpA
20033 Desio (Milan), Italy

8.3 H. Schafer
Kraftwerk Union - Karlstein Labs
D-8520 Erlangen 2
Federal Republic of Germany

8.4 T. Mumford
Weir Pumps Ltd
Glasgow, U.K.

8.5 P. Grison
Electricite de France
Direction des Etudes et Recherches
Chatou, France

8.6 R.J. Gilbert
Dept. des Etudes Mecaniques et Thermiques
Centre d'Etudes Nuclearie de Saclay
Gif-sur-Yvette, France

(Responses are shown in Appendix E)

APPENDIX B

Pump Design Data

(From References t.5 - t.15, Appendix D, Selected Bibliography)

PUMP DESIGN DATA

TABLE OF CONTENTS

a. DARLINGTON NGS 'A' PUMP/MOTOR SET
  a.1 Background Data
  a.2 Characteristics
  a.3 Detailed Data
  a.4 Figures
    a.4.1 Cross-Section
    a.4.2 Containment Penetration
    a.4.3 Gland Seal Internals
    a.4.4 Piping Layout
    a.4.5 Characteristic Curve

b. BRUCE NGS 'A' PUMP/MOTOR SET
  b.1 Background Data
  b.2 Characteristics
  b.3 Detailed Data
  b.4 Instrumentation and Annunciation
  b.5 Figures
    b.5.1 Cross-Section
    b.5.2 Support
    b.5.3 Seals and Bushing
    b.5.4 Characteristic Curve

c. PICKERING NGS 'B' PUMP/MOTOR SET
  c.1 Background Data
  c.2 Characteristics
  c.3 Detailed Data
  c.4 Figures
    c.4.1 Cross-Section
    c.4.2 Schematic including Instrumentation and Auxiliary Circuits
    c.4.3 Plant Layout
    c.4.4 Characteristic Curve
    c.4.5 Characteristic Curve under Varying Conditions

TABLE OF CONTENTS (Cont'd)

d. PT. LEPREAU & G-2 PUMP/MOTOR SET
  d.1 Background Data
  d.2 Characteristics
  d.3 Detailed Data
  d.4 Instrumentation and Annunciation
  d.5 Figures
    d.5.1 Cross-Section
    d.5.2 Reactor Layout
    d.5.3 Characteristic Curve

a. DARLINGTON NGS 'A' PUMP/MOTOR SET (From Bib. t. 10 & 11)

a.1 Background Data

Number : 4 of 4

Rated flow per pump : 3.1 m^3/s

Rated head : 224.2 m (D_2O)

Suction pressure : 9.6 MPa (absolute)

Operating temperature : 255.2°C

Operating load (hot) : 6.7 MWe

Operating load (cold) : 8.4 MWe

Design pressure : 12.9 MPa (absolute)

Design temperature : 279.4°C

Rotating inertia : 1500 kg . m^2

Outlet Quality : 2%

a.2 Characteristics

- vertical, single stage, single suction, double discharge, centrifugal;
- casing inside containment with top flange supported at containment structure;
- removable shaft coupling connects motor to pump;
- squirrel cage induction motor, totally enclosed, water cooled, 13.8 KV supply voltage;

- inertia of 15 Kg m^2 provided by high inertia rotor (rundown matches normal reactor power decay characteristic);

- brake to stop unpowered pump;

- gland seal cooling provided by heat transport feed pump discharge; backup supply from either heat transport pump discharge or shutdown cooling pump discharge;

- pump also designed to run at 65% reactor power and flow rate in series with unpowered, heated pump;

- pumps may be required to operate in cavitation;

- no automatic trip of pump set due to heat transport system process perturbations or due to loss of services to pumps; electrical overload protection trip provided;

- designed and manufactured for Class 1 components in CSA CAN3-N285.0-M81; also: Class 1, Section III of ASME Boiler & Press Vessel Code;

- three main end-face mechancial seals and one back-up seal; each seal capable of operating with full system pressure differential.

a.3 Detailed Data

Rated Operating Conditions

Fluid	Heavy water (D_2O)	
Temperature	266.2°C	(511°F)
Flow per pump	3.1 m^3/sec	(40915 IGPM)
Head	224.2 m	(735 ft)
Suction Pressure	9.61 MPa(a)	(1394 psia)
Discharge Pressure	11.41 MPa(a)	(1654 psia)
Efficiency (pump)	89%	
Motor Operating Load		

Cold	8.4 MW	(11270 HP)
Hot	6.7 MW	(9000 HP)
Inertia (combined)	1500 $kg.m^2$	(35600 $lbm\text{-}ft^2$)

Design Conditions

Temperature	279°C	(535°F)
Pressure	12.75 MPa(a)	(1850 psia)

Materials

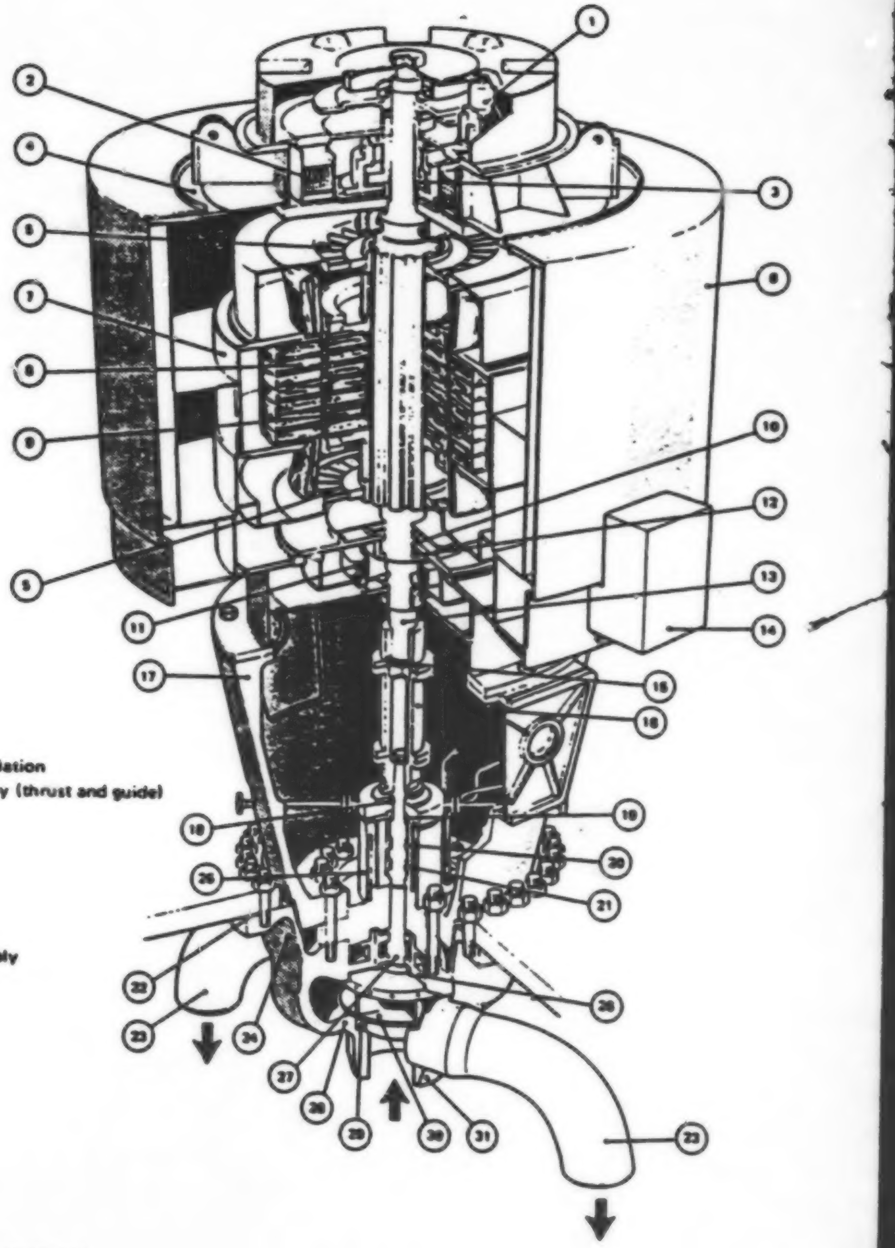
Case	SA-216, WCC
Cover	SA-350, LF2
Case Wear Ring	ASTM A-473, TP420
Impeller	SA-487, CA-6NM
Shaft	SA-479, TP410
Shaft Sleeve	ASTM 1-296, TP CA-15

Connections

Inlet	Carbon Steel 609.6 mm (24") OD Schedule 100 pipe
Outlet	Carbon Steel 533.4 mm (21") OD 443 mm (17.44") ID

Motors

Quantity	4
Type	Squirrel Cage Induction
Power Requirement	13.2 KV, 3 phase, 50 Hz
Shaft Speed	1800 rpm
Rotor Inertia	1450 kg m^2
Capacity (rated)	9400 KW (12,600 hp)
Specification	NK38-TH33130-01



LEGEND

1. Brake
2. Non-drive end oil cooler installation
3. Non-drive end bearing assembly (thrust and guide)
4. Motor hood
5. Fan
6. Motor housing
7. Stator housing
8. Stator core assembly
9. Rotor core assembly
10. Drive end guide bearing assembly
11. Oil cooler installation
12. Labryinth seal
13. Motor shaft
14. Main terminal box
15. Thrust disc
16. Spacer coupling
17. Motor mount
18. Back-up seal
19. Tertiary mechanical seal
20. Secondary mechanical seal
21. Primary mechanical seal
22. Containment seal gaskets
23. Pump discharge elbow; (double discharge)
24. Pump support
25. Pump cover
26. Pump bearing
27. Pump shaft
28. Pump case
29. Pump impeller/case wear ring
30. Pump impeller
31. Suction spool piece

FIGURE a.4.1

CROSS-SECTION

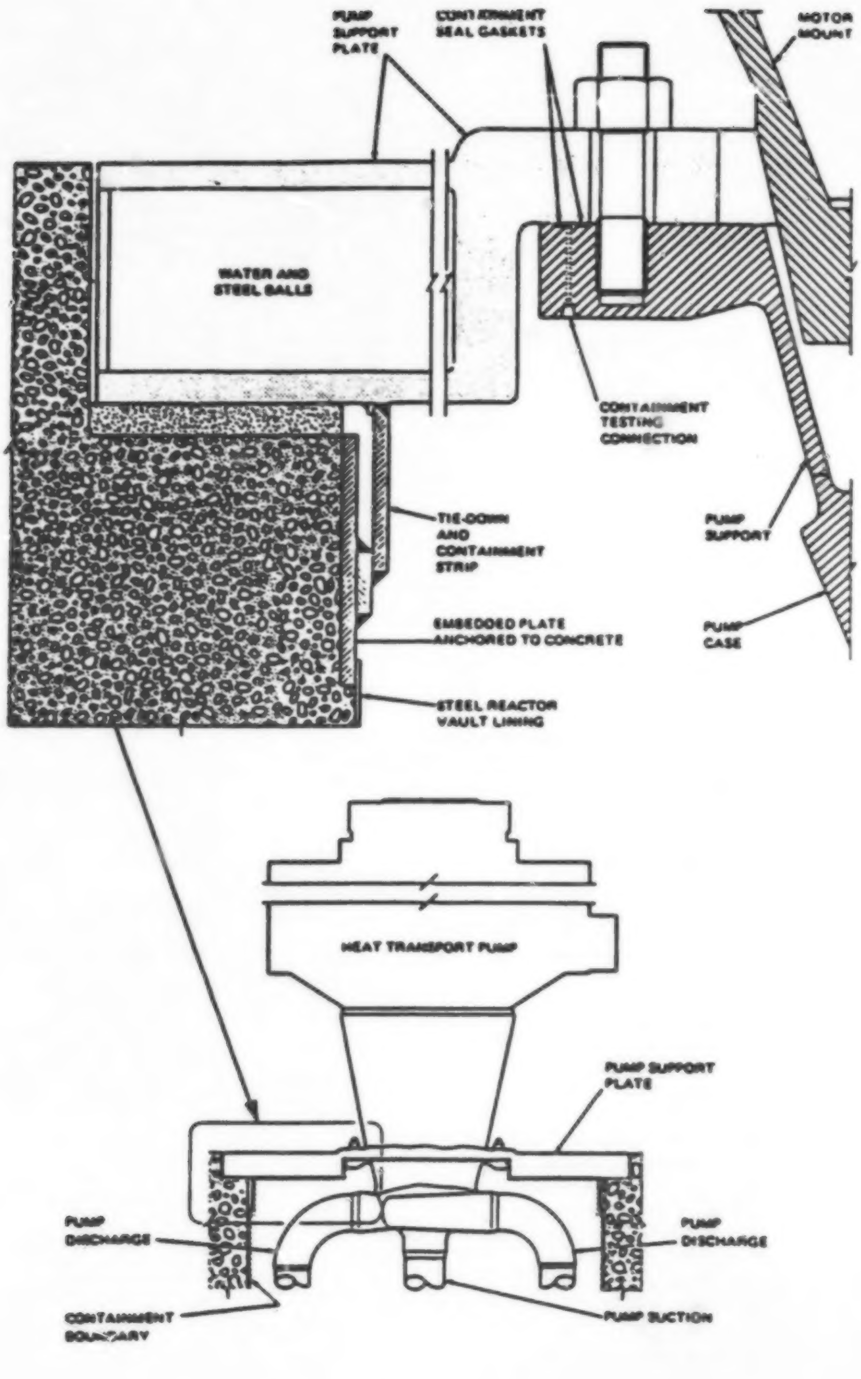

PUMP SUPPORT PLATE
CONTAINMENT SEAL GASKETS
MOTOR MOUNT
WATER AND STEEL BALLS
CONTAINMENT TESTING CONNECTION
TIE-DOWN AND CONTAINMENT STRIP
PUMP SUPPORT
EMBEDDED PLATE ANCHORED TO CONCRETE
PUMP CASE
STEEL REACTOR VAULT LINING
HEAT TRANSPORT PUMP
PUMP SUPPORT PLATE
PUMP DISCHARGE
PUMP DISCHARGE
CONTAINMENT BOUNDARY
PUMP SUCTION

FIGURE a.4.2.

CONTAINMENT PENETRATION

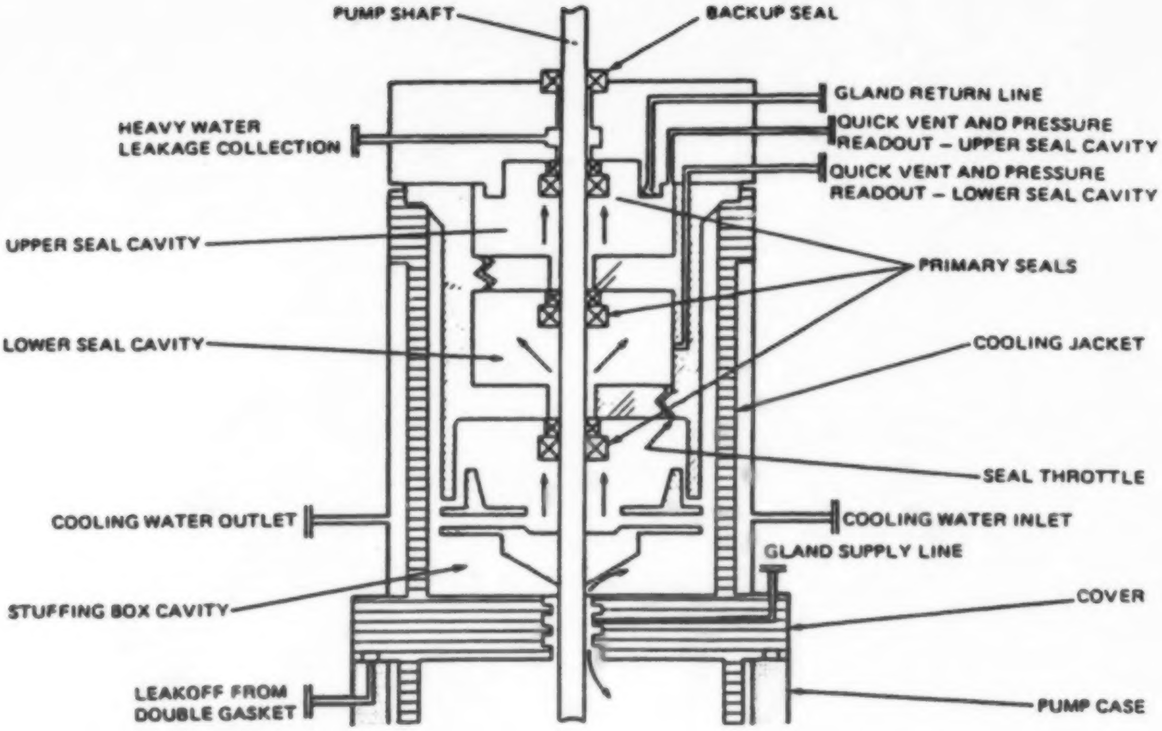
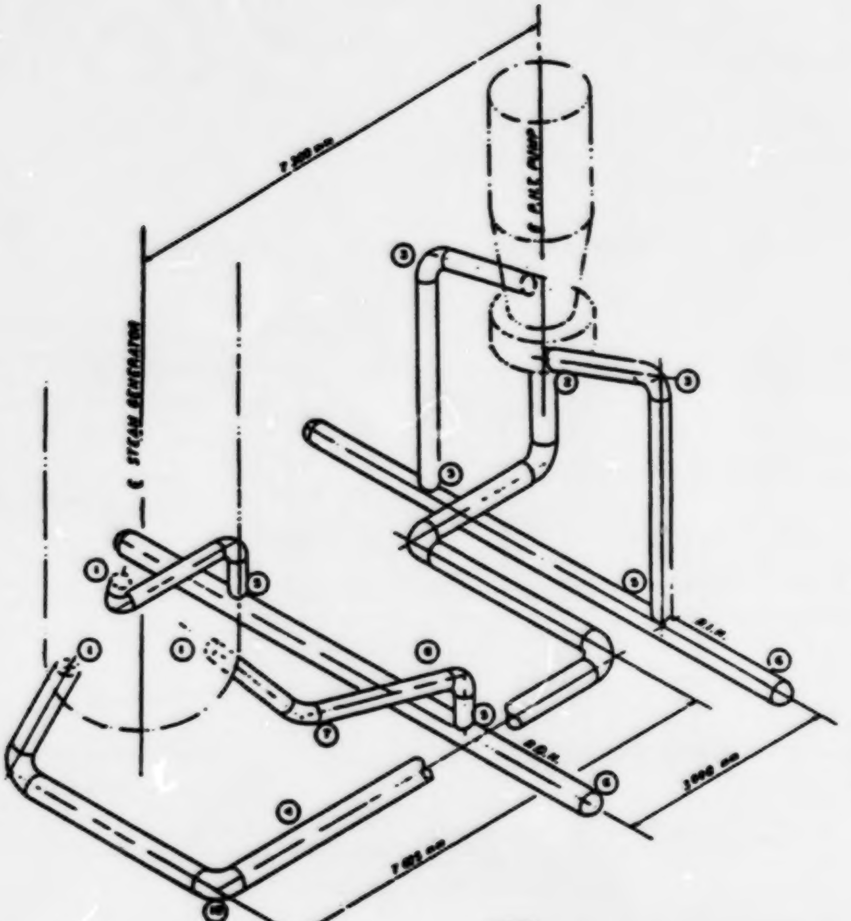

PUMP SHAFT
BACKUP SEAL
GLAND RETURN LINE
HEAVY WATER LEAKAGE COLLECTION
QUICK VENT AND PRESSURE READOUT – UPPER SEAL CAVITY
QUICK VENT AND PRESSURE READOUT – LOWER SEAL CAVITY
UPPER SEAL CAVITY
PRIMARY SEALS
LOWER SEAL CAVITY
COOLING JACKET
SEAL THROTTLE
COOLING WATER OUTLET
COOLING WATER INLET
GLAND SUPPLY LINE
STUFFING BOX CAVITY
COVER
LEAKOFF FROM DOUBLE GASKET
PUMP CASE

FIGURE a.4.3.

GLAND SEAL INTERNALS


7 200 mm
℄ R.H.T. PUMP
℄ STEAM GENERATOR
R.I.H.
R.O.H.
1 900 mm
①
②
③
④
⑤
⑥
⑦
⑧
⑩

NOTES

1 ELEVATION OF S.G. NOZZLES CONNECTION 115 202 mm
2 WELD LINE OF SUCTION PIPE TO R.H.T. PUMP EL. N 114 830 mm
3 WELD LINE OF DISCHARGE PIPE TO R.H.T. PUMP ELEVATION N. 114 736 mm
4 SUCTION LINE EL. 113 780 mm
5 a BOTH R.I.H. & R.O.H. LENGTH 8580 mm*
b CONNECTION OF BOTH RISER PIPES TO R.O.H & PUMP DISCHARGE PIPES TO R.I.H. ARE APPR. HEADERS ¼ POINTS
6 HEADERS EL. 112 170 mm (ROH), EL. 111 790 mm (RIH)
7 TRUE ANGLE :113°
8 TRUE ANGLE : 96°
9 PIPE & HEADER SIZES: AS PER TABLE 1.
10 ALL ELBOWS: LONG RADIUS
* NOTE: NOT INCLUDING END CAPS.

FIGURE a.4.4

PIPING LAYOUT

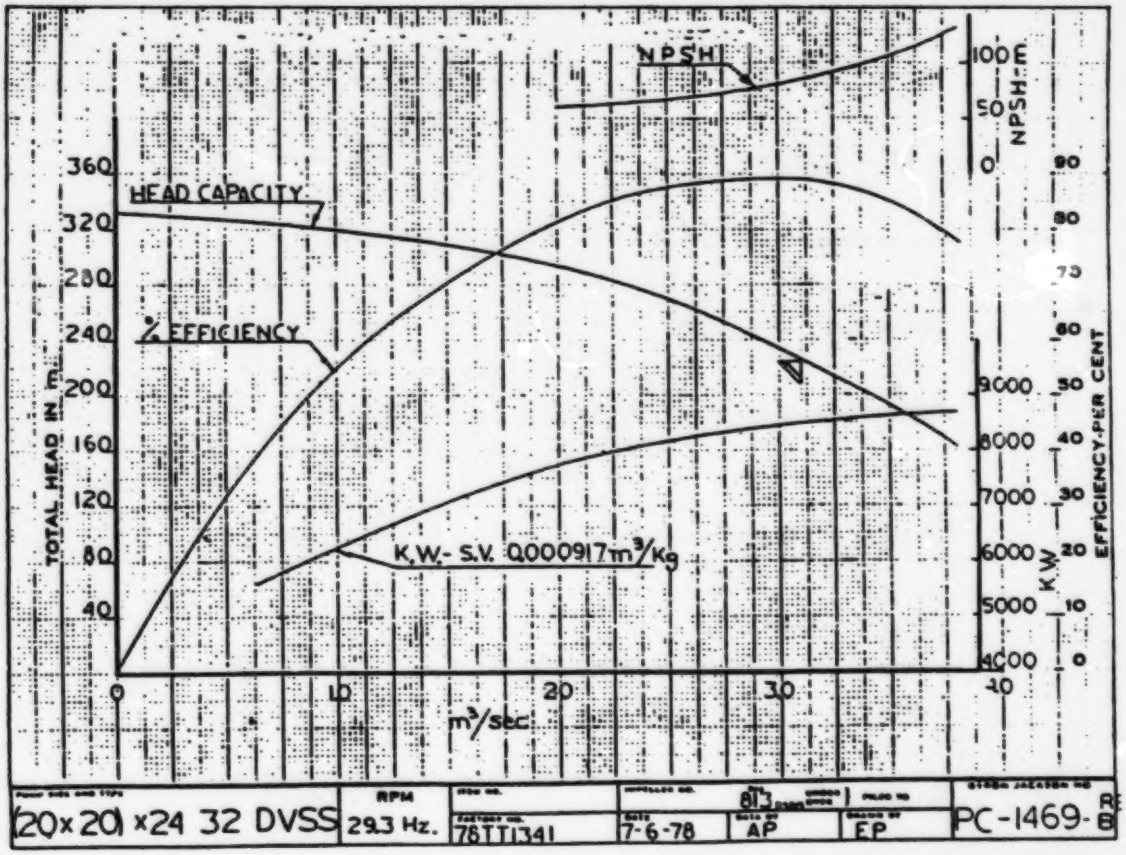


FIGURE a.4.5.

CHARACTERISTIC CURVE

b. BRUCE NGS 'B' PUMP/MOTOR SET (From Bib. t. 7, 8, & 9)

b.1 Background Data

Number : 4 of 4

Rated flow per pump : 3,300 L/s

Rated head : 213 m (D_2O)

Suction pressure : 8.83 MPa (g)

Operating temperature : 265°C

Operating load (hot) : 6.8 MWe

Operating load (rated/cold) : 8.6 MWe
: 26

Design discharge pressure (hot) : 11.3 MPa (g)

Design temperature : 274°C

Pump/Motor assembly inertia : 2,107 kg.m^2

Outlet Quality : Subcooled

b.2 Characteristics

- vertical, single suction, double volute;
- bellows-type seal connects top of pump casing to containment liner;
- larger pump feet (than Bruce 'A') due to higher postulated burst pipe loads;
- double mechanical face seals, supplied with filtered gland cooling water, backed by tertiary axial segmented-graphite bushing-type

seal; drain pipe from cavity between mechanical seals and backup seal; seals instrumented for failure detection;

- pump gland houses seal cartridge comprising three basic assemblies: primary mechanical seal, secondary mechanical seal, radial contact backup seal; interseal pressure and secondary seal leakage monitored;

- primary seal seals against 8.89 MPa (g) pressure, with 4.45 MPa (g) back pressure ; secondary seal seals against 4.45 MPa (g) with close to atmospheric backpressure ; each seal capable of operating at full system pressure for a short time;

- upper guide & thrust bearings combined in housing at top of motor; guide bearing tilting pads contoured to provide greater centering force on shaft; pre-set pad clearance; improved oil circulation through bearing cavity eliminating restriction on reverse speed; (improved from Bruce 'A');

- differential pressure across hydrostatic bearing maintained by auxiliary impeller located in upper shroud of main impeller;

- bearing pressurizing flow leaks from pockets past glands to drain gutters; flow from top gland returns to back hub of impeller, results in pressure at bottom end of throttle bushing slightly above pump suction pressure.

- auxiliary impellers assure adequate flow to hydrostatic bearings in turbining conditions (change from Bruce 'A');

- vertical water/air cooled squirrel cage induction type motor; variable speed drive;

- conventional two-winding transformer;

- braking permitted from 720 rpm, forward or reverse direction; located at upper end of motor shaft for maintenance; (improved from Bruce 'A', similar to 600 MWe);

- motor power 6.6 KV, 3-phase, 60 hertz supplied to motor from individual two winding, silicon filled transformers; transformers fed from 13.8 KV power system;

- 8 rectangular pockets in stator, each receiving pressurized D_2O via titanium alloy restricting orifice from reservoir surrounding the bearing; reservoir supplied from pump discharge;

- inertia of motor rotating assembly sufficient to provide adequate rundown circulation on loss of power to pump following ractor trip; ensures establishment of thermosiphoning;

- rotational inertia obtained using laminated packages of stainless steel punchings mounted outboard of rotor core; (solid flywheel in Bruce 'A');

- pump mfr : Byron Jackson;

- motor mfr: Canadian General Electric;

- ASME Code, Section III, Class 1 Nuclear Pumps.

b.3 Detailed Data

Pumps

Pump Shaft Seals:	Face type mechanical seal
Primary	Face type mechanical seal
Secondary	Face type mechanical seal
Tertiary	Axial segmented-ring graphite

Dimensions:

Impeller	28 1/2" (724 mm)
Inlet	22" (560 mm)
Outlet	22" (560 mm)

Arrangement	Connected in parallel, two pumps for each bank of boilers
Material:	
Casing	ASTM A216-Grade WCC
Wear ring	ASTM A296 Gr. Ca 40
Impeller	ASTM A487 CA6NM
Shaft	ASTM A479 TP410
Weight (each pump & motor or set)	153,300 lb (69.5 Mg)
Impeller diameter, max.	31.5 in (800.1 mm)
Shaft diameter, nominal	8 in (203 mm)
Normal flow per pump	43,600 Igpm (3.3 m^3/s)
Total head	700 ft (210 m)
Suction pressure	1281 psig (8.84 MPa (g))
NPSH, available (normal operation)	1220 ft (370 m)
NPSH, requied (normal operation)	250 ft (47 m)
Design pressure - Design 1	1700 psig (11.72 MPa (g))
- Design 2	1635 psig (11.27 MPa (g))
Design temperature - Design 1	100°F (38°C)
- Design 2	526°F (274°C)
Suction & discharge connections	22 in. inside diameter pipe
Fluid pumped:	D_2O
Fluid pH	9.5 to 10.5
Fluid temperature	509°F (265°C)
Fluid specific gravity	0.856
Main Drive Motors:	
Type	Squirrel cage, induction, air-water cooled
Synchronous speed	1800 rpm
Power requirement	6.6 kV, 3 phase, 60 hertz
Shaft horsepower, rated	

conditions:

Hot system, approx. 9,000 hp (6.7 MW)
Cold system, approx. 11,000 hp (8.2 MW)

b.4 INSTRUMENTATION AND ANNUNCIATION

Pump/Motor Set Variables for CRT Display

- Phase "A" Winding Temperature
- Phase "B" Winding Temperature
- Phase "C" Winding Temperature
- Up - thrust Bearing Temperature
- Down - thrust Bearing Temperature
- Upper Guide Bearing Temperature
- Lower Guide Bearing Temperature
- Thrust Bearing Oil Temperature
- Lower Guide Bearing Oil Temperature
- Motor Cooling Air Temperature
- Pump Speed

Pump/Motor Set Vibration Signals for CRT Display & Annunciaion

- Motor top bearing velocity
- Motor bottom bearing velocity
- Half coupling displacement (2x)
- Motor top bearing displacement (2x)
- Motor bottom bearing displacement (2x)
- Motor top shaft run-out

PHT System, Pump/Motor Set Vibration Alarms

- Top bearing vibration high
- Bottom bearing vibration high
- Half coupling vibration high

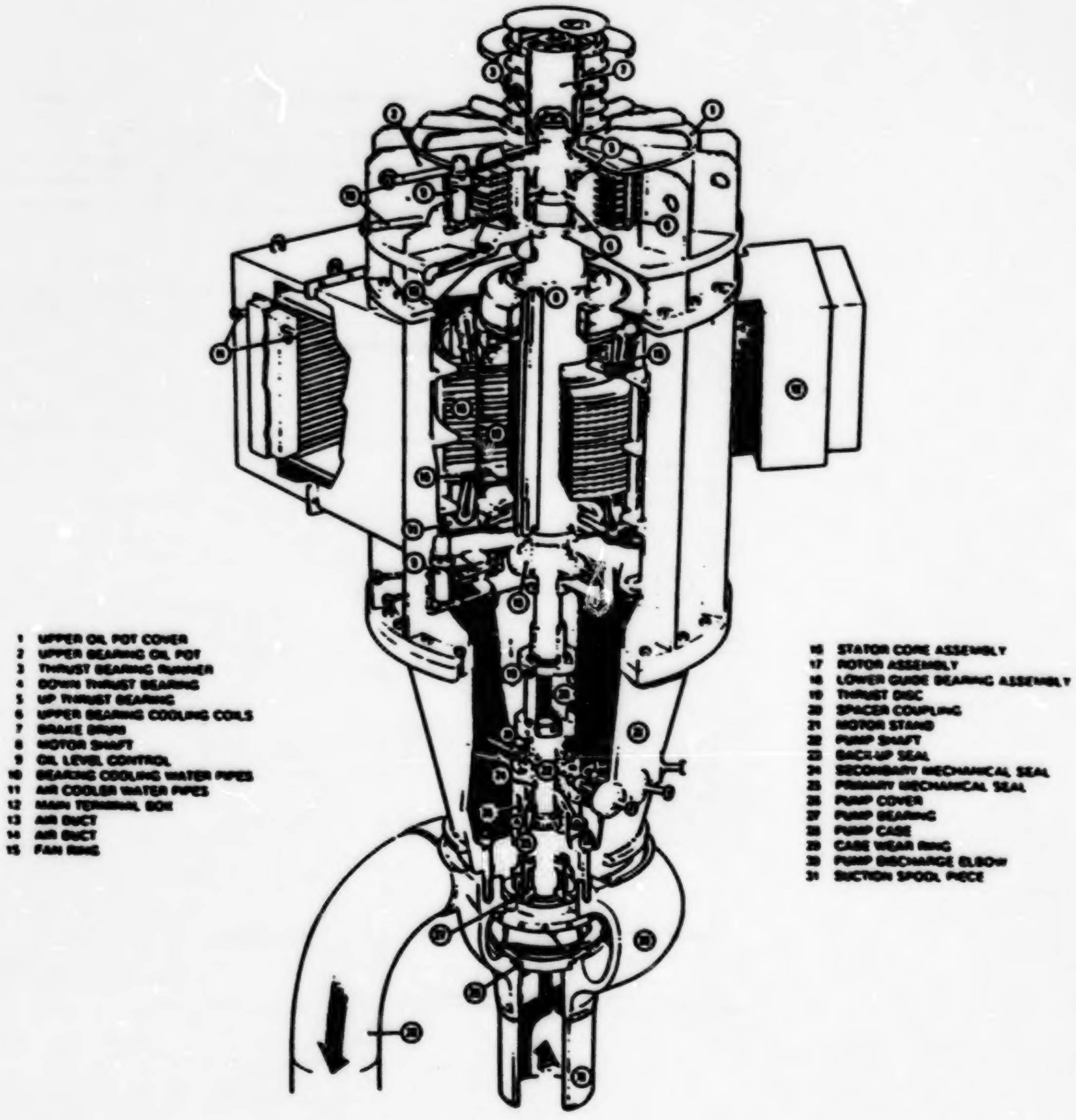

1 UPPER OIL POT COVER
2 UPPER BEARING OIL POT
3 THRUST BEARING RUNNER
4 DOWN THRUST BEARING
5 UP THRUST BEARING
6 UPPER BEARING COOLING COILS
7 BRAKE DRUM
8 MOTOR SHAFT
9 OIL LEVEL CONTROL
10 BEARING COOLING WATER PIPES
11 AIR COOLER WATER PIPES
12 MAIN TERMINAL BOX
13 AIR DUCT
14 AIR DUCT
15 FAN RING
16 STATOR CORE ASSEMBLY
17 ROTOR ASSEMBLY
18 LOWER GUIDE BEARING ASSEMBLY
19 THRUST DISC
20 SPACER COUPLING
21 MOTOR STAND
22 PUMP SHAFT
23 BACK-UP SEAL
24 SECONDARY MECHANICAL SEAL
25 PRIMARY MECHANICAL SEAL
26 PUMP COVER
27 PUMP BEARING
28 PUMP CASE
29 CASE WEAR RING
30 PUMP DISCHARGE ELBOW
31 SUCTION SPOOL PIECE

FIGURE b.5.1.

CROSS-SECTION

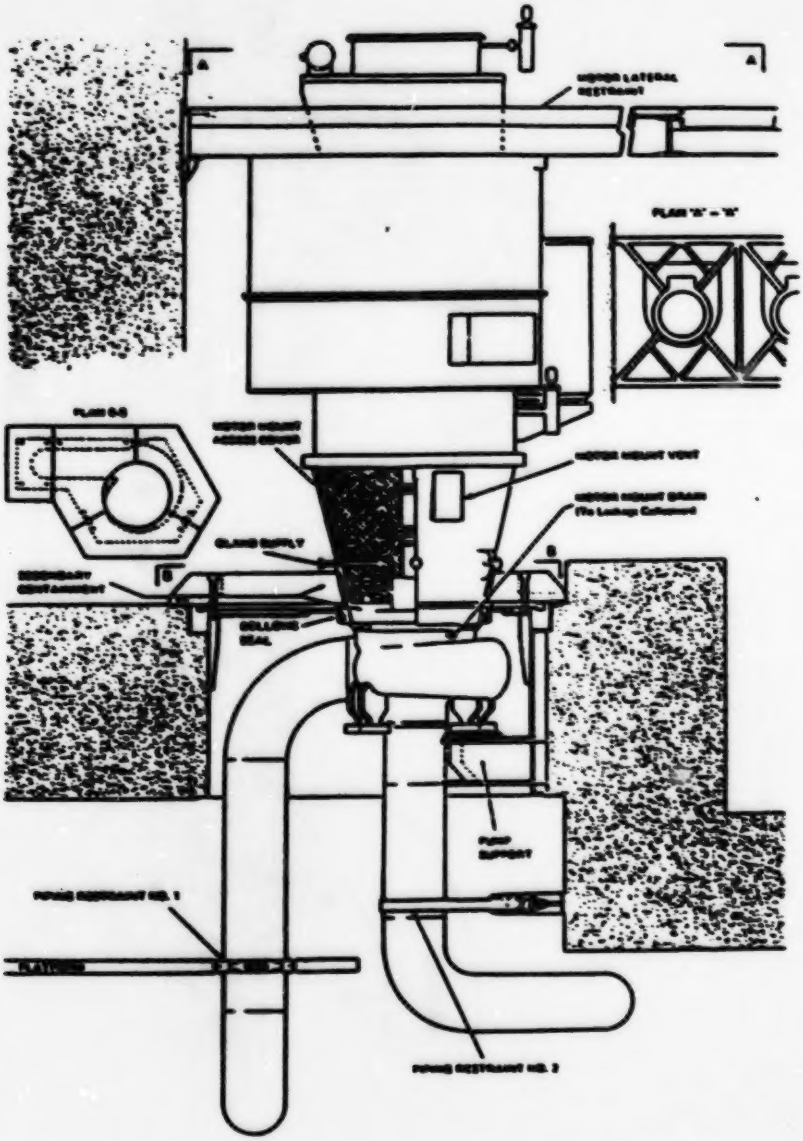

A
A
MOTOR LATERAL RESTRAINT
PLAN "A" - "A"
PLAN B-B
MOTOR MOUNT ACCESS COVER
MOTOR MOUNT VENT
MOTOR MOUNT DRAIN (To Leakage Collection)
GLAND SUPPLY
B
B
SECONDARY CONTAINMENT
BELLOWS SEAL
PUMP SUPPORT
PIPING RESTRAINT NO. 1
PLATFORM
PIPING RESTRAINT NO. 2

FIGURE b.5.2

SUPPORT

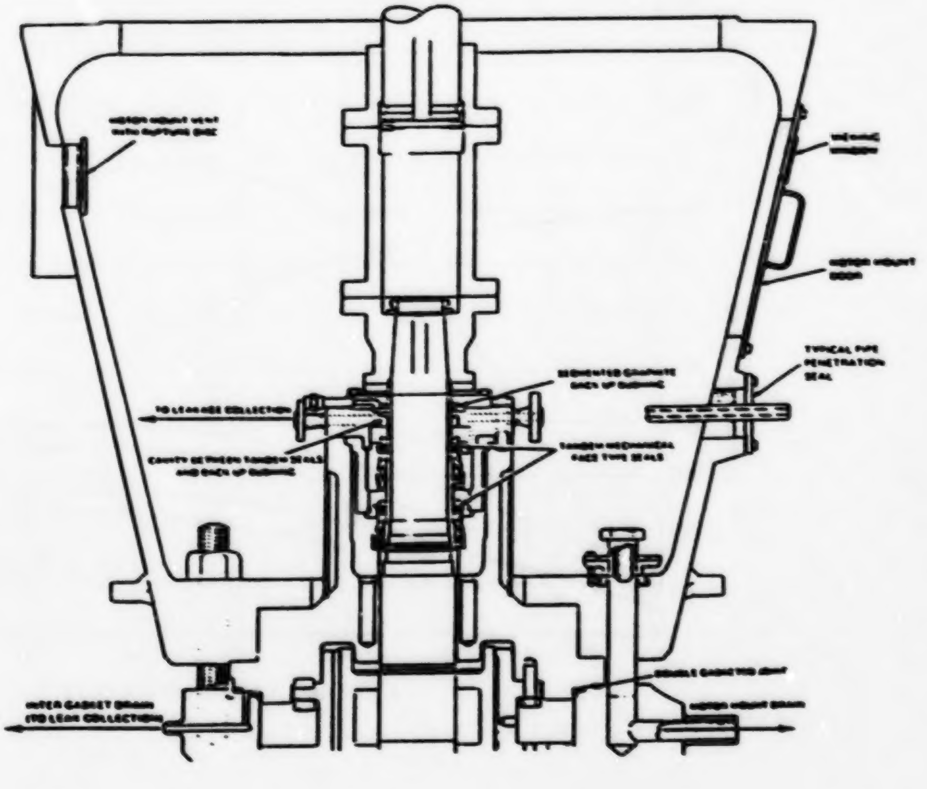

MOTOR MOUNT VENT
WITH RUPTURE DISC
VIEWING
WINDOW
MOTOR MOUNT
DOOR
TYPICAL PIPE
PENETRATION
SEAL
SEGMENTED GRAPHITE
BACK UP BUSHING
TO LEAKAGE COLLECTION
CAVITY BETWEEN TANDEM SEALS
AND BACK UP BUSHING
TANDEM MECHANICAL
FACE TYPE SEALS
DOUBLE GASKETED JOINT
MOTOR MOUNT DRAIN
INTER GASKET DRAIN
(TO LEAK COLLECTION)

FIGURE b.5.3

SEALS AND BUSHINGS

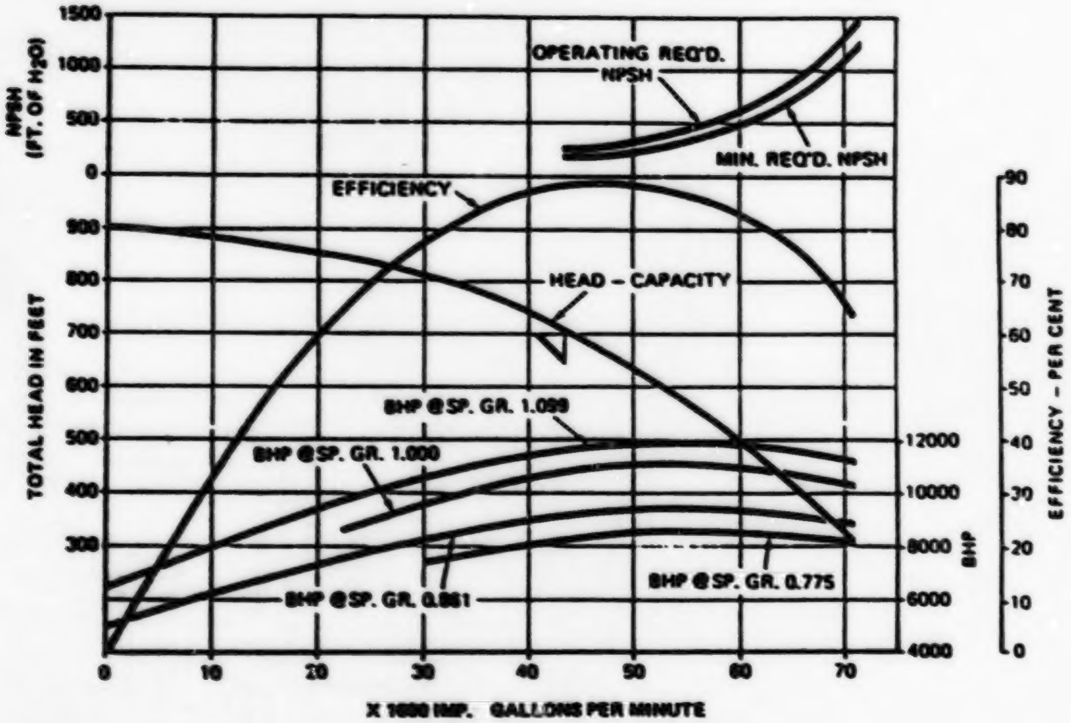


FIGURE b.5.4

CHARACTERISTIC CURVE

c. PICKERING NGS 'B' (From Bib. t. 5 & 6)

c.1 Background Data

Number	: 12 of 16
Rated flow per pump	: 0.765 m^3/s
Rated head	: 143 m (D_2O)
Suction pressure	: 8.48 MPa (guage)
Operating load	: 1.17 MW
Design pressure	: 11.1 MPa (guage)
Design temperature	: 265°C
Total rotating inertia	: 657 kg. m^2
Outlet quality	: Subcooled

c.2 Characteristics

- vertical, single stage, single suction, single discharge, centrifugal;
- vertical bottom suction, horizontal discharge nozzle;
- cover/stuffing box mounted on top of casing, sealed by double spiral wound "Flexitallic" gaskets; within stuffing box are thermal barrior, throttle bushing, D_2O lubricated pump bearing, shaft seals & shaft;
- shaft seals: throttle bushing, two mechanical seals, vapour containment seal; both seals can operate at full system pressure, one housed in removable cartridge; vapour seal at top of stuffing box, consists of labyrith air bushing and graphite axial bushing;

- gland seal flow from one of two pressurizing pumps; small portion of flow passes into pump casing through radial throttle bushing;

- vertical, totally enclosed, water-cooled squirrel cage induction motor;

- 80% of weight of total pump/motor set supported by two spring-mounted hanger rods attached to lugs at upper flange of motor mount; remaining 20% of weight supported by sliding support under pump suction tee;

- removal shaft coupling connects motor & pump;

- motor flywheel prolongs pump operation after loss of motor power;

- upper & lower guide bearings and upper thrust bearing on motor; motor bearings lubricated by cooled oil;

- class 1, Section III, ASME Boiler and Pressure Vessel Code;

- pump suction 14" dia. disharge 14" dia. 24" max. impeller diametr, carbon steel.

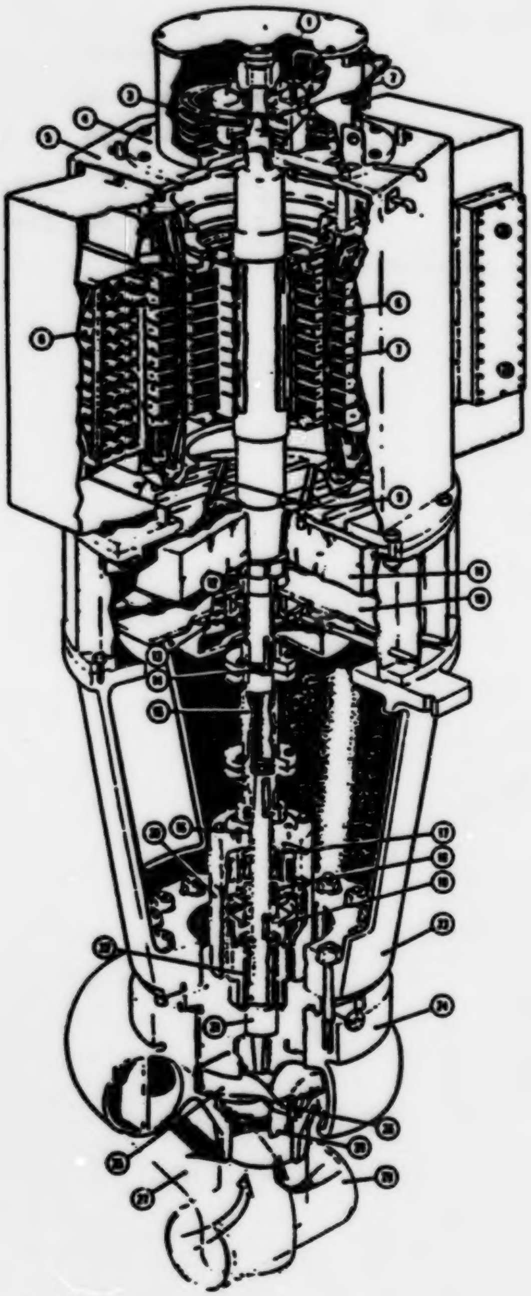
c.3 Detailed Data

Manufacturer:	Byron Jackson of Canada Ltd.
Model Number:	14 x 14 x 24 DVSS
Pump Rated Capacity:	765 L/s (10,100 Igpm)
Pump Rated Head:	480 ft D_2O
Pump Efficiency at Rated Conditions:	85%
Pump Speed:	1796
Suction Pressure (approx):	8274 kPa (1200 psig)
Temperature at Suction:	235°C (480°F)
Temperature of Gland Supply:	38°C (10°F)
Gland Supply Flow:	0.75 L/s (10 Igpm)
Temperature Rise of Gland Flow:	-12°C (10°F)

Leakage Through Seal Cartridge:	20 cc/min max.
Heavy Water Holdup:	209 dm^3 (7.4 ft^3)
Weight of Pump Casing:	2585 kg (5,700 lbs)
Weight of Pump(dry):	9072 kg (20,000 lbs)
Weight of Pump Motor Assembly (dry):	19 504 kg (43,000 lbs)
Moment of Inertia of Flywheel:	615 kg/m^2 (14,600 lb-ft^2)
Total Moment of Inertia of Rotating Parts of Pump & Motor:	657 kg/m^2 (15,600 lb-ft^2)
Minimum flow through pump to prevent overheating when operating at rated speed against shutoff head:	95 L/s (1250 Igpm)
Flow to leakage collection with failed mechanical seal:	0.75 L/s (1250 Igpm)
First Critical Speed of rotating parts including motor:	2524 rpm
Pump Jacket Cooling Water Flow:	0.75 L/s (10 Igpm)
Impeller Diameter	
Maximum:	611 mm (24 inches)
Trimmed:	584 mm (23 inches)
Pump Design Pressure:	11.1 MPa (g) (1600 psig)
Pump Design Temperature:	265°C (510°F)
Manufacturer:	Canadian General Electric
Model Number:	K-7946-V
Nameplate Rating:	1900 HP
Normal Operating Power (3-3 Pumps, 480°F):	1575 HP
Maximum Power (2-0 Pumps, 100°F - 515°F):	2145 HP
Maximum Power (2-2 Pumps, 470°F - 515°F)	1830 HP
Synchronous Speed:	1800rpm
Full Load Speed:	1791 rpm
Enclosure:	Tewac
Rated Volts/Phase/Frequency:	4000/3/60
Insulation Class:	B
Service Factor:	1.15
Ambient Temperature:	40°C (104°F)

Temperature Rise at	
1900 HP:	**70°C (158°F)**
1575 HP:	**55° (131°F)**
2145 HP:	**85°C (185°F)**
1830 HP:	**65°C (149°F)**
Hot Spot Temperature at	
1146 HP:	**125°C (257°F)**
Full Load Current at	
1900 HP:	**245 amps**
1575 HP:	**206 amps**
2145 HP:	**275 amps**
1830 HP:	**237 amps**
Locked Rotor Current:	**1586 amps**
Locked Rotor Power Factor:	**18.83%**
Locked Rotor KVA:	**10986**
Motor Efficiency at	**closed valve**
Full Load:	**95.2%**
75% Load:	**95%**
50% Load:	**93.7%**
Power Factor at	
Full Load:	**87.5%**
75% Load:	**87%**
50% Load:	**82%**
Torque - % of Nameplate FL Rating	**closed valve**
Locked Rotor:	**85%**
Pulled up:	**85%**
Breakdown:	**220%**
Rated FL Torque:	**7552 Nm (5570 ft-lb)**
Cooling Water, at 68°F:	
Stator:	**4 L/s (50 Igpm)**
Bearings:	**0.45 L/s (6 Igpm)**
Winding RTD Number:	**3**
Bearing RTD Number:	**3**
Space heater Volts/Amps/Phase:	**550/200/1**
Recommended Brg Lub at 100°F:	**300 SSU**
Bearings	
Thrust:	**Kingsbury BVB**
Guide:	**two 5 shoe Kingsbury**

First Critical Speed of Motor: 2340 rpm
Max Run-out at Coupling
Horizontal: 0.0015 inches
Vertical: 0.0150 inches
Weight of Motor (complete): 10 432 kg (23,000 lbs)
Weight of Rotor plus Flywheel: 3883 kg (8,650 lbs)
Weight of Flywheel: 2722 kg (6,000 lbs)
Acceleration Time to Rated Speed: 19 seconds



1 UPPER GUIDE BEARING
2 THRUST BEARING
3 BEARING COOLING COILS
4 UPPER BEARING BRACKET
5 MOTOR FRAME
6 ROTOR
7 STATOR
8 STATOR COOLERS
9 SPACE HEATERS
10 LOWER BEARING BRACKET
11 FLYWHEEL
12 LOWER GUIDE BEARING
13 BEARING COOLING COILS
14 THRUST DISC
15 SPACE COUPLING
16 VAPOUR CONTAINMENT SEAL
17 SECONDARY MECHANICAL SEAL
18 PRIMARY MECHANICAL SEAL
19 RECIRCULATION IMPELLER
20 PUMP COVER
21 THROTTLE BUSHING
22 PUMP BEARING
23 MOTOR STAND
24 PUMP CASE
25 PUMP IMPELLER
26 CASE WEAR RING
27 PUMP DISCHARGE ELBOW
28 ANTI-ROTATION VANE
29 SUCTION TEE

FIGURE c.4.1

CROSS-SECTION

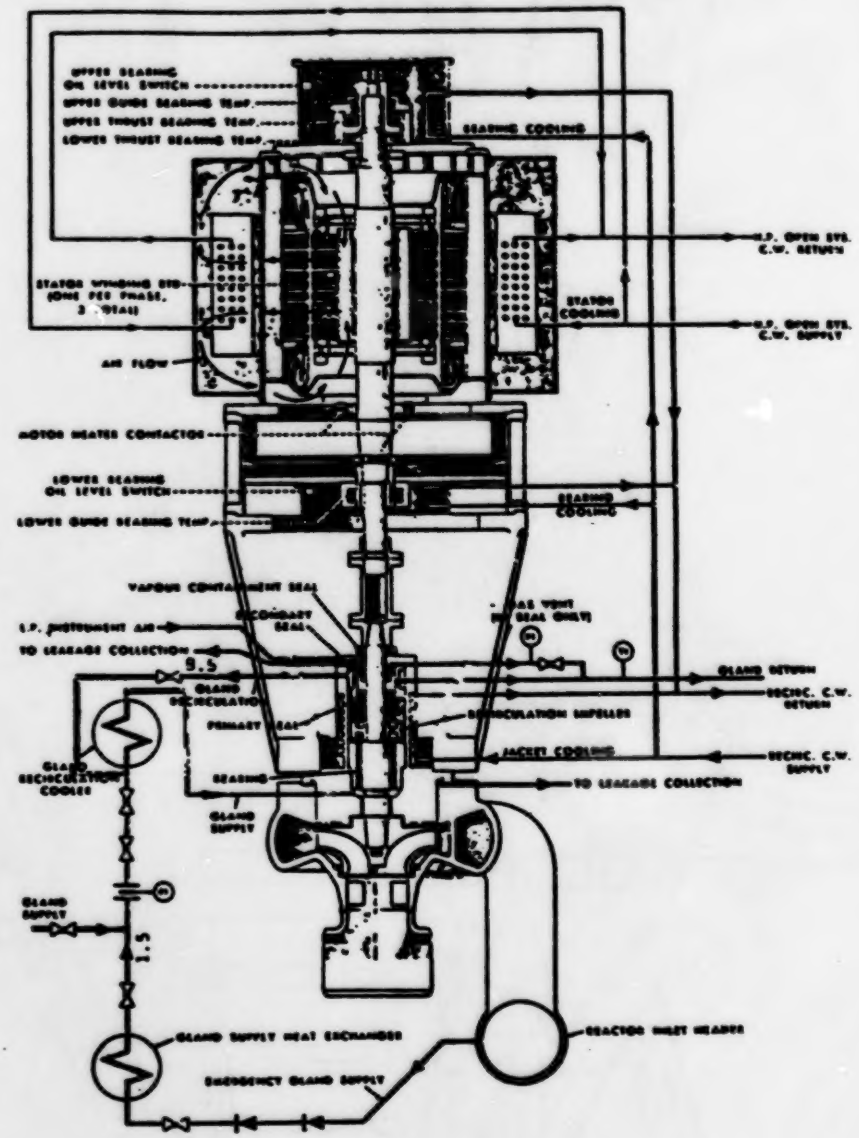

UPPER BEARING OIL LEVEL SWITCH
UPPER GUIDE BEARING TEMP.
UPPER THRUST BEARING TEMP.
LOWER THRUST BEARING TEMP.
BEARING COOLING
STATOR WINDING RTD (ONE PER PHASE, 3 TOTAL)
AIR FLOW
STATOR COOLING
L.P. OPEN SYS. C.W. RETURN
L.P. OPEN SYS. C.W. SUPPLY
MOTOR HEATER CONTACTOR
LOWER BEARING OIL LEVEL SWITCH
LOWER GUIDE BEARING TEMP.
BEARING COOLING
VAPOUR CONTAINMENT SEAL
SECONDARY SEAL
GAS VENT (NO SEAL ONLY)
L.P. INSTRUMENT AIR
TO LEAKAGE COLLECTION
9.5
GLAND RECIRCULATION
GLAND RETURN
RECIRC. C.W. RETURN
PRIMARY SEAL
RECIRCULATION IMPELLER
JACKET COOLING
RECIRC. C.W. SUPPLY
GLAND RECIRCULATION COOLER
BEARING
TO LEAKAGE COLLECTION
GLAND SUPPLY
GLAND SUPPLY
1.5
GLAND SUPPLY HEAT EXCHANGER
REACTOR INLET HEADER
EMERGENCY GLAND SUPPLY

FIGURE c.4.2

SCHEMATIC INCLUDING
INSTRUMENTATION AND
AUXILIARY CIRCUITS

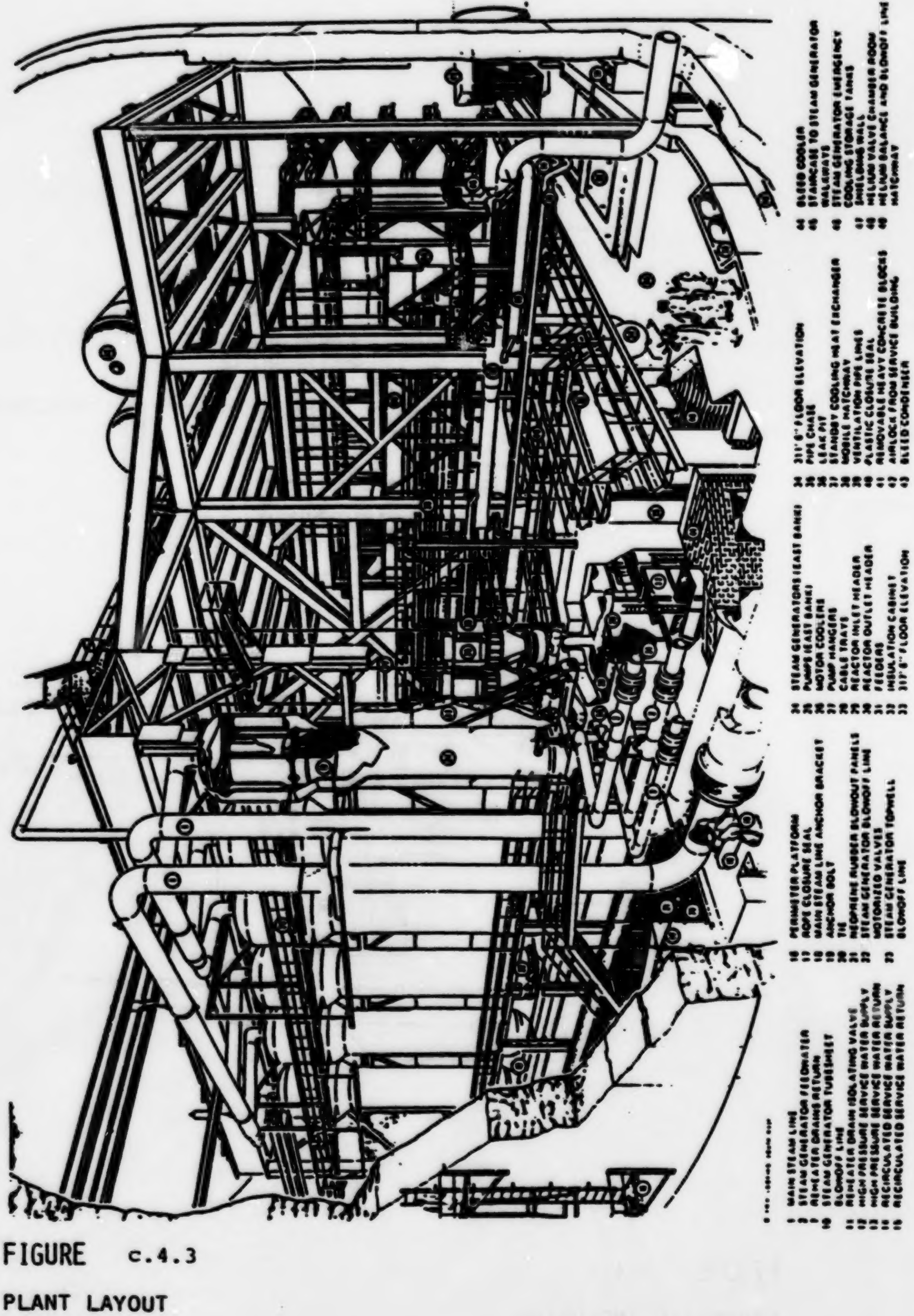

1 MAIN STEAM LINE
2 STEAM GENERATOR FEEDWATER
3 REHEATER DRAINS RETURN
10 STEAM GENERATOR TUBESHEET
BLOWOFF LINE
11 REHEATER DRAIN ISOLATING VALVE
12 HIGH PRESSURE SERVICE WATER SUPPLY
13 HIGH PRESSURE SERVICE WATER RETURN
14 RECIRCULATED SERVICE WATER SUPPLY
15 RECIRCULATED SERVICE WATER RETURN
16 PERIMETER PLATFORM
17 ROPE CLOSURE SEAL
18 MAIN STEAM LINE ANCHOR BRACKET
19 ANCHOR BOLT
20 TIE
21 NEOPRENE RUBBER BLOWOUT PANELS
22 STEAM GENERATOR BLOWOFF LINE
MOTORIZED VALVES
23 STEAM GENERATOR TOPWELL
BLOWOFF LINE
24 STEAM GENERATORS (EAST BANK)
25 PUMPS (EAST BANK)
26 MOTOR COOLERS
27 PUMP HANGERS
28 CABLE TRAYS
29 REACTOR INLET HEADER
30 REACTOR OUTLET HEADER
31 FEEDERS
32 INSULATION CABINET
33 311' 6" FLOOR ELEVATION
34 311' 6" FLOOR ELEVATION
35 PIPE CHASE
36 LEAK PIT
37 STANDBY COOLING HEAT EXCHANGER
38 MOBILE HATCHWAY
39 VENTILATION PIPE LINES
40 PLASTIC CLOSURE SEAL
41 REMOVABLE HEAVY CONCRETE BLOCKS
42 AIRLOCK FROM SERVICE BUILDING
43 BLEED CONDENSER
44 BLEED COOLER
45 STAIRCASE TO STEAM GENERATOR WALKWAYS
46 STEAM GENERATOR EMERGENCY COOLING STORAGE TANKS
47 SHIELDING WALL
48 HELIUM VALVE CHAMBER ROOM
49 HELIUM BALANCE AND BLOWOFF LINE
HATCHWAY

FIGURE c.4.3

PLANT LAYOUT

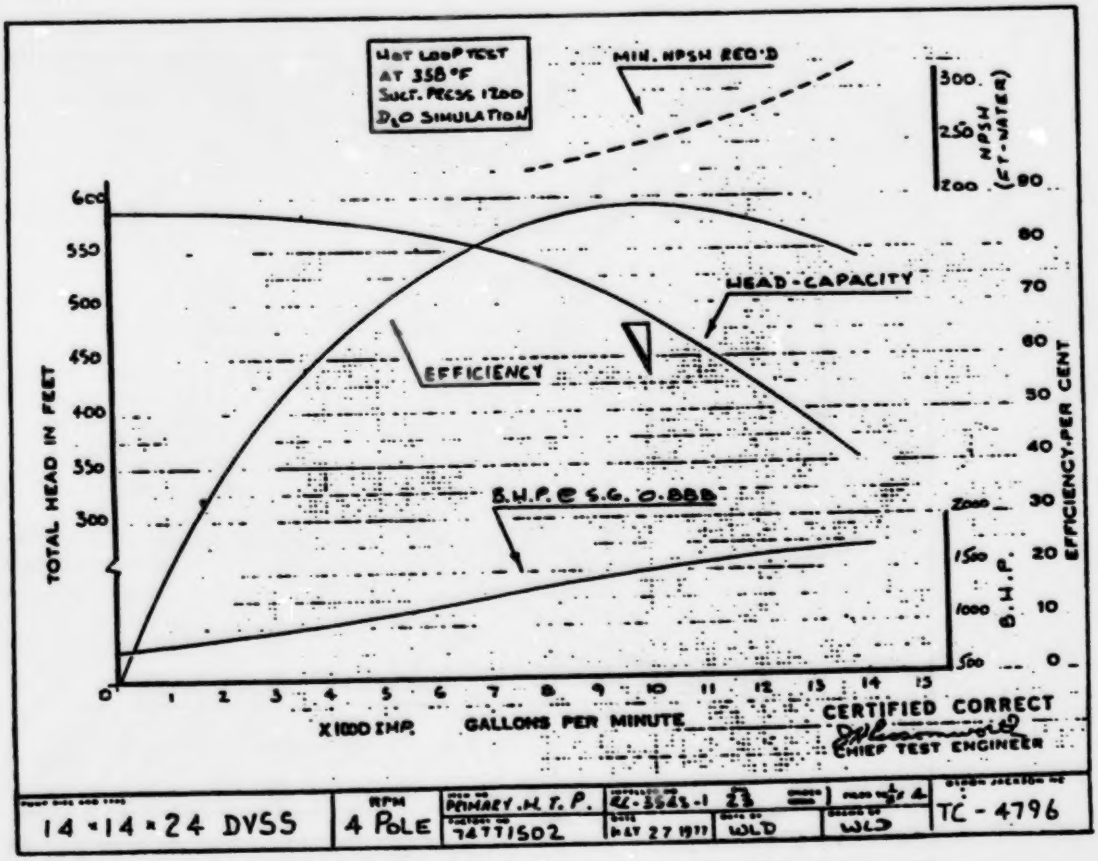


FIGURE c.4.4

CHARACTERISTIC CURVE

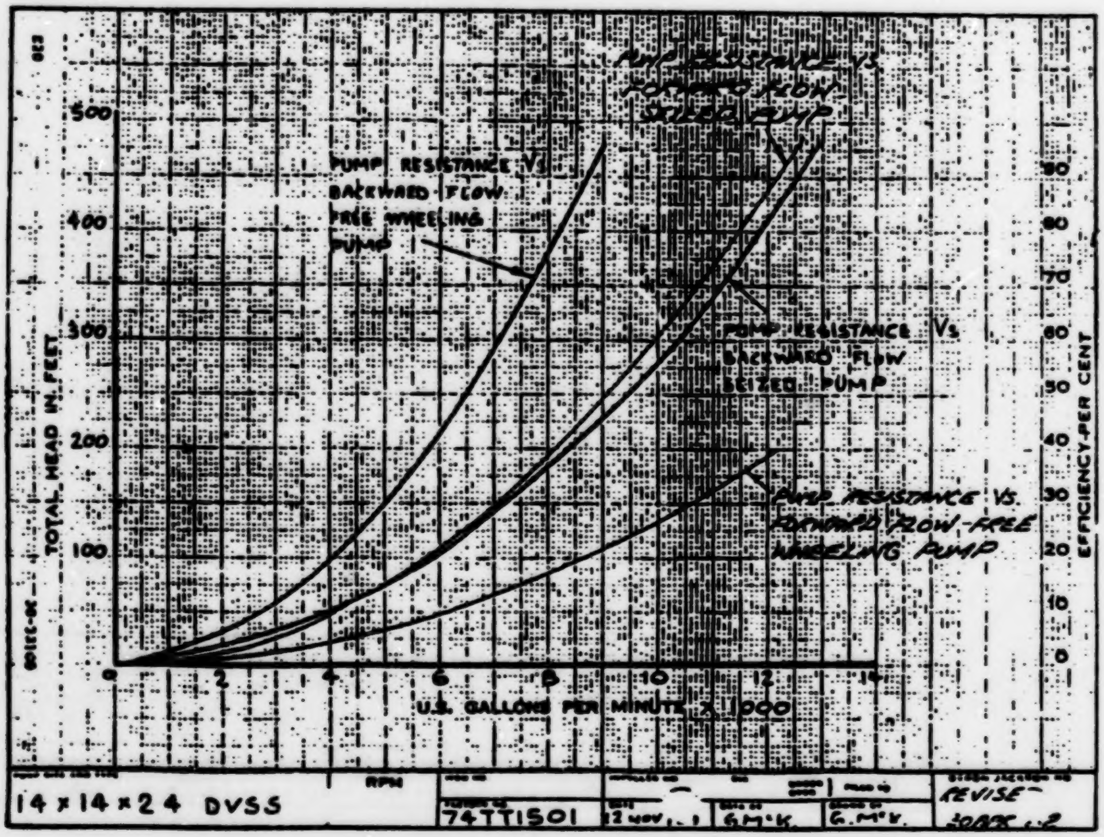


FIGURE c.4.5

CHARACTERISTIC CURVE
UNDER VARYING CONDITIONS

d. PT LEPREAU & G-2 PUMP MOTOR SET (From Bib. t. 12-14)

d.1 Background Data

Number : 4 of 4

Rated flow per pump : 2,228 l/s

Rated head : 215 m (d_2O)

Suction pressure : 9.5 MPa

Operating temperature : 266°C

Operating load : 4.78 MWe

Design pressure : 12.9 MPa (g)

Design temperature : 279°C

Motor Rotating inertia : 1264 $kg.m^2$

Outlet quality : 4%

d.2 Characteristics

- vertical, single stage, double volute, single suction, double discharge, centrifugal;

- pump cover mounted on top of casing with double gasket seal, creating pressure boundary; cover consists of bottom flange which rates with casing and vertical cylindrical section (gland);

- three identical high pressure shift seals in series, with one backup; seal consists of rotor and stator; seal in parallel with throttle (controls gland flow); small flow through rotor-stator interface, most flow through throttle;

- each seal can operate at full differential pressure;
- gland seal supply from discharge feed pumps, backup from primary pump discharge;
- motor: vertical, totally enclosed, air-water cooled, squirrel cage, induction motor;
- pump supported by motor mount, which is bolted to top of pump casing;
- motors supported by spring hangers and seismic restraints ; pumps supported by headers through the connected piping system;
- seismic restraint pads located on motor casing; steel starts running from bldg. concrete centred over pads to carry horizontal loads resulting from earthquake;
- shielding between pump casing and motor;
- removable coupling between motor & pump;
- motor rotational inertia sufficient to prolong pump rundown after loss of power, approximate match with reactor power rundown;
- motor has two removable radial bearings and double acting thrust bearing, lubricated by cool, pressurized oil;
- rotor bearing - water lubricated guide bearing in pump; tilting pad type, oil-lubricated radial guide bearing at top & bottom of drive motor;
- axial thrust (up or down) taken by double acting tilting pad type thrust bearing at top of motor;
- brake capable of stopping unpowered pump subjected to 80% forward flow;

- pump and piping thermally insulated by feeder cabinet and insulation on casing;

- volute casing is a casting; two symmetrical, tangential, discharge nozzles, terminating in elbows, connected to headers below ; double discharge advantageous mechanically for mfg, produces better burst pipe safety analysis situation.

- 16" dia. discharge (x2); 20" dia. suction;

- max. peak-to-peak amplitude of vibration on pump or motor: 0.003" (horizontally or vertically), normal operation;

- first critical speed of complete rotating assembly not less than 130% of rated speed.

d.3 Detailed Data

Pumps

Rated Operating Conditions		
Fluid	Heavy Water	
Temperature	266°C	(511°F)
Flow	2228 l/s	(29,400 Igpm)
Head	215 m	(705 ft)
Suction Pressure		
Normal	9.50 MPa	(1378 psia)
Minimum	1.38 MPa	(200 psia)
Discharge Pressure	11.22 MPa	(1627 psia)
Efficiency (Pump)	88%	
Design Conditions		
Temperature	279°C	(535°F)
Pressure	12.9 MPa (g)	(1870 psig)
Motor Operating Load	4.78 MWe	(6407 HP)

Materials	
Case	Carbon Steel SA216 GRWCC
Cover	Stainless Steel SA336 GRF8C
Case Wear Ring	Stainless Steel DIN 1.4313 (CA-6NM)
Impeller	Stainless Steel DIN 1.4313 (CA-6NM)
Shaft	Stainless Steel DIN 1.4313 (CA-6NM)
Shaft Sleeve	Stainless Steel ASTM A473 Type 431
Connections	
Inlet	Butt welded 20 inch Sch. 100
Outlet	Butt welded 2 x 16 inch Sch. 100
Pump Motor	
Quantity	4
Type	Squirrel Cage Induction
Shaft Speed	1800 RPM
Rotor Inertia	1264 kg m^2 (30,000 lb ft^2)
Capacity	6.7 MWE (9000 HP)
Code Classification	ASME Section III Class 1
Seismic Classification	DBE Category A

d.4 Instrumentation & Annunciation

Temperature Monitoring

- stator winding for each phase of motor
- motor upthrust bearing
- motor downthrust bearing
- motor upper guide bearing
- motor lower guide bearing
- oil in upper guide & thrust bearing reservoir
- oil in lower guide bearing reservoir

Level Monitoring

- bearing oil reservoirs (upper & lower)
- air cooler, water leakage

Vibration Monitoring

- upper motor guide bearing - velocity
- lower motor guide bearing - velocity
- pump bearing
- pump-motor coupling, pump end - displacement

Speed Monitoring

Motor Current

- current transducer on one phase

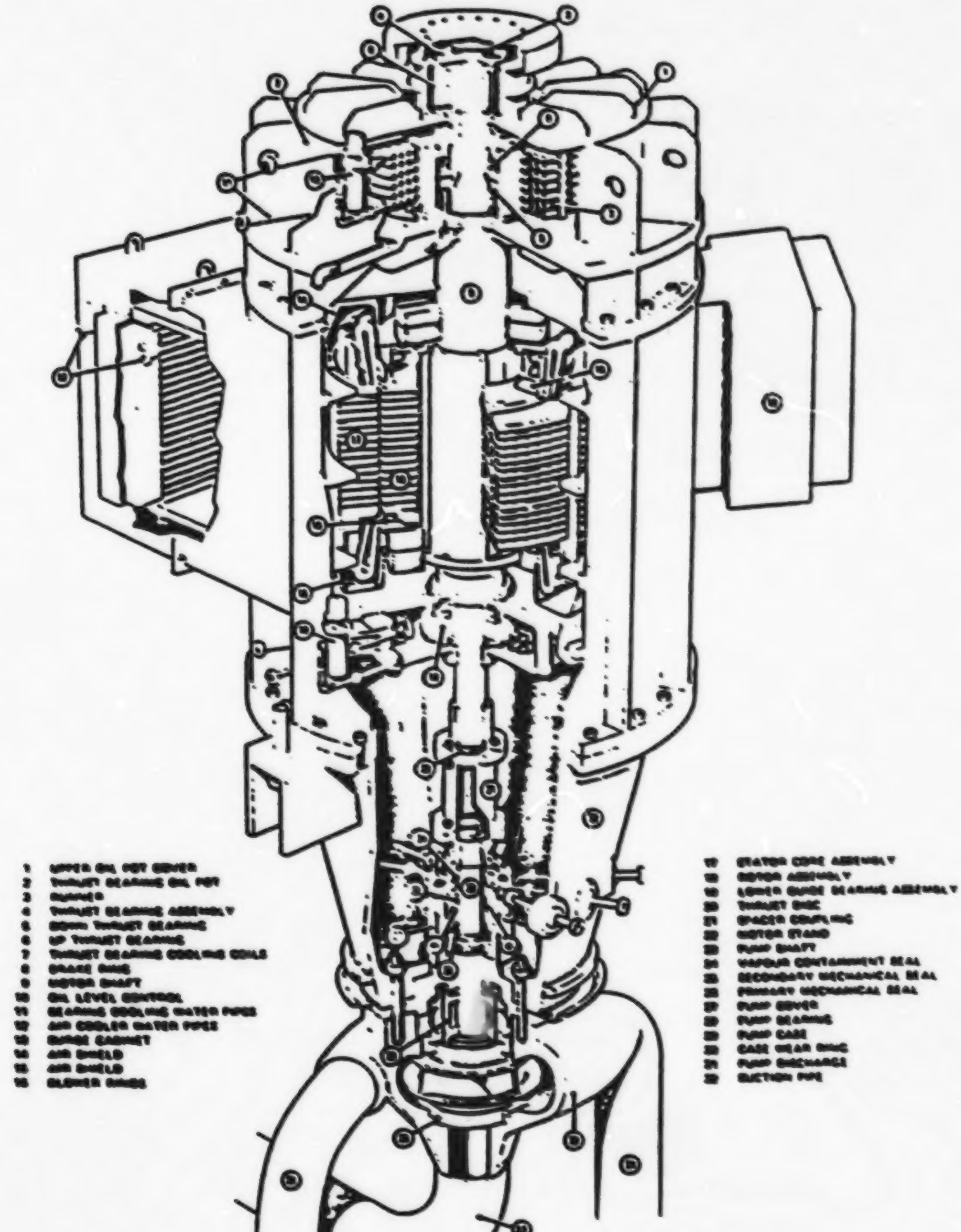

1 UPPER OIL POT COVER
2 THRUST BEARING OIL POT
3 RUNNER
4 THRUST BEARING ASSEMBLY
5 DOWN THRUST BEARING
6 UP THRUST BEARING
7 THRUST BEARING COOLING COILS
8 BRAKE RING
9 MOTOR SHAFT
10 OIL LEVEL CONTROL
11 BEARING COOLING WATER PIPES
12 AIR COOLER WATER PIPES
13 SURGE CABINET
14 AIR SHIELD
15 AIR SHIELD
16 BLOWER RINGS
17 STATOR CORE ASSEMBLY
18 ROTOR ASSEMBLY
19 LOWER GUIDE BEARING ASSEMBLY
20 THRUST DISC
21 SPACER COUPLING
22 MOTOR STAND
23 PUMP SHAFT
24 VAPOUR CONTAINMENT SEAL
25 SECONDARY MECHANICAL SEAL
26 PRIMARY MECHANICAL SEAL
27 PUMP COVER
28 PUMP BEARING
29 PUMP CASE
30 CASE WEAR RING
31 PUMP DISCHARGE
32 SUCTION PIPE

FIGURE d.5.1

CROSS-SECTION

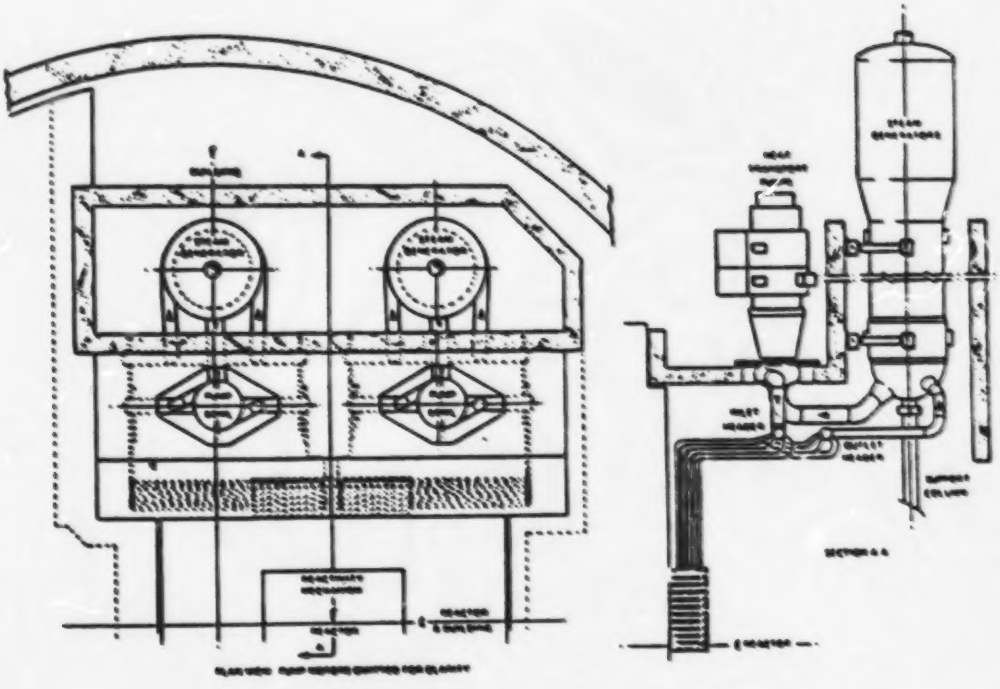

STEAM GENERATORS
HEAT TRANSPORT PUMP
INLET HEADER
OUTLET HEADER
SUPPORT COLUMN
SECTION A-A

FIGURE d.5.2

REACTOR LAYOUT

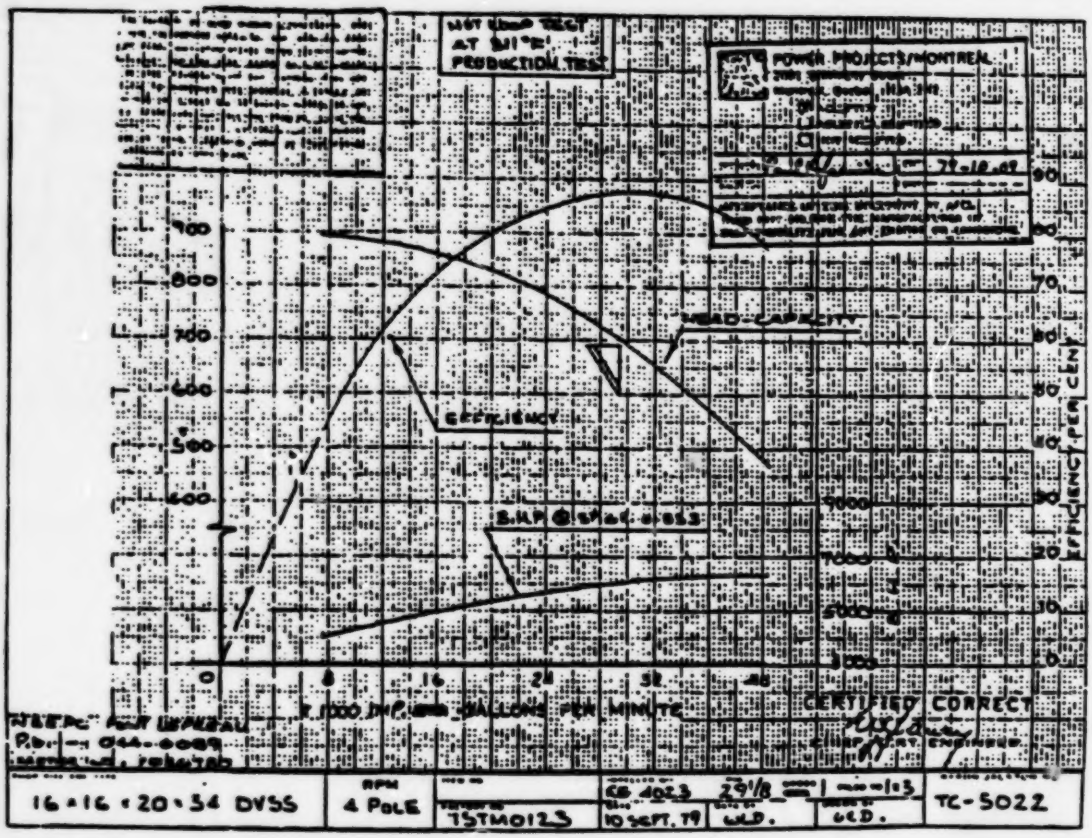


FIGURE d.5.3

CHARACTERISTIC CURVE

APPENDIX C

Ontario Hydro Visit Report

Date : February 13, 14, 1984

Davis Engineering Representative : B M. Guthrie

Ontario Hydro Contacts :		
	B.R. Churchill	Nuclear Systems Dept.
	R.G. Lyons *	Nuclear Systems Dept.
	P.R. Burchett	Nuclear Systems Dept.
	R. Hartlan	Research Division
	M. Boyd	Research Division
	D.K. Vijay	Mechanical Design Dept.
	T. Stevens	Mechanical Design Dept.

Activities:

- informal discussions with contacts above;
- inspecton of Kipling Pump Test Complex;
- brief search of library;
- reviewing and gathering reports, memos, paper, etc.

General Comments:

- Hydro personnel were open and willing to discuss problems or answer questions;
- An extensive amount of information was gathered, especially reports (see attachment). Little time was available for technical questions. The information gathered could be considered only a brief summary of the total amount of work that has been completed;
- The Pump Test Facility work has a perceived commercial value. This is Ontario Hydro's major concern with respect to confidentiality and their reason for requesting a review of the draft final report;

- Operations was not involved in the discussions as the Nuclear Systems Department has already obtained all relevant information;

- A literature survey was performed two or three years ago, with no significant findings. Hydro considers themselves the world experts in the field;

- Several possible contacts were obtained - i.e. individuals or groups who have expressed interest in Hydro's work;

- The initial, primary objective of the test program was to investigate the pumps mechanical/structure integrity under LOCA conditions at the pump;

- Testing and analysis of the Bruce and Pickering pumpsets have been completed. The methodology and the results are considered satisfactory;

- The two-phase/pump vibration question is recognized as very complex. Limitations to the analytical and testing procedures are recognized. Perceived areas of difficulty include the excitation of pressure pulses, and piping analysis models;

- Testing and analyses have become more sophisticated with experience. Earlier work (i.e. pre-1983, Bruce) involved estimated values and assumptions which have since been refined.

Thermal-hydraulics

- Pressure pulse velocity measurement is difficult. Pressure measurements for straight lengths of pipe are not applicable in complex geometries. Initial assumptions of pulse propogation at flow velocities (Bruce) were changed in later analyses (Pickering) to sonic velocities.

---

* Primary/Formal Contact

- The void fraction at the pump is estimated from the loop average void, which is calculated from the holding tank volume. Measuring devices have proven unreliable.

- Because of the loop's smaller size and compact layout, the Pickering tests include a full blowdown and ECI simulation. The Bruce loop was limited to low void fractions due to the loop size and geometry.

Mechanical

- The pumps have survived the testing remarkably well and Ontario Hydro has confidence in their mechanical integrity during periods of vibration.

- The only mechanical damage to the pumps during testing was some scoring on a thrust bearing. The seals are not preceived as a problem.

- Structural/vibration models of the pumpsets have not been carried out. The tests are perceived as proof of the operating requirement.

Piping

- Most work is done by the Mechanical Design Department using the computer code NASTRAN. However, the Nuclear Systems Department has also developed a customized program.

- The pump is a very important component in the pipe vibration analysis.

- Piping analyses are largely based on theory using Pump Test Facility results as background data. However, the Test Facility was not set up for this purpose. In future work, redesign of the analytic model, or possible scale models would enhance results.

- Improvements could be made in the tests for piping analyses by refining the temperature transients, pipe geometry and data (post) processing from tests.

Test Loop

- Differences and characteristics of the loops are summarized below:

- Bruce :
  - . testing limited by vibration in hairpin piping loops, not the pumpset;
  - . testing also limited by the undersized vapour holding tank;
  - . motor frame was not supported, but was instrumented (anchored in the station);
  - . BJ CAN III seals used;

- Pickering :
  - . full blowdown/ECI capability;
  - . donut shaped piping instead of header; concrete foundation added for stability;
  - . arbitrary placement of pipe supports (ex. - support of elbow on discharge piping);

- Darlington :
  - . pump is similar to Bruce, slightly higher performance, double (rather than single) discharge;
  - . cone motor mount is part of containment

ontario hydro

700 University Avenue, Toronto, Ontario M5G 1X6

1984 02 27

OUR FILES: RGL Diary Files

Davis Engineering
Consulting Engineers
Attention: Brian M. Guthrie, P. Eng.
Project Manager
1481 Cyrville Road
OTTAWA, Ontario
K1B 3L7

Dear Mr. Guthrie:

Assessment of Primary Heat Transport Pump Integrity During Operation Under LOCA Conditions

Enclosed please find the five category 'H' reports you requested during your recent visit and which required a Director's approval for release. The report numbers are listed as an attachment to this letter. Also attached, for record purposes is a list of the reports which were given to you while you were here.

I understand the S.E.R. you requested has been submitted to the AECB, I suggest you contact them for a copy.

I would like to wish you success on your project and look forward to receiving your draft report.

RG Lyon

R.G. Lyons
Supervising Design Engineer
Nuclear Equipment Unit
Nuclear Equipment and Processes Section
Nuclear Systems Department
(H10 A11, Telephone: 416-592-4924)

Enc

LIST OF REPORTS

REPORTS ENCLOSED:

Research Report No.

80-426-H
82-77-H
82-239-H
B83-64-H
83-483-H

REPORTS PREVIOUSLY RECEIVED:

Research Report No.

80-236-K
81-45-K
82-25-K
81-502-K
82-413-K

Design and Development Report No.

80262
82194
82197
82236
83292
83328
83410

RGL:ab
84-02-27

APPENDIX D

Selected Bibliography

APPENDIX D : TABLE OF CONTENTS

List of Keywords

a. Ontario Hydro, Design & Development Division
b. Ontario Hydro, Research Division
c. AECL
d. Canada - Operations
e. U.S. - Pump/Flow
f. U.S. - Flow (Only)
g. U.S. - Pump/Mechanical/Vibration/Seals/Motors
h. U.S. - Piping (General)
i. U.S. - Vibration (General)
j. U.S. - Seismic
k. International - Operating
l. France
m. Germany
n. Japan
o. Italy
p. U.K.
q. Eastern Block Countries
r. Miscellaneous Countries
s. Computer Codes
t. Background References

LIST OF KEYWORDS USED

Nuclear

Pumps

Centrifugal

Seals

Piping

Motor

Electrical

Cavitation

Two-Phase

Fluid Forces

Mech. Design

Vibration

Siesmic

Analysis

Testing

Operations

Monitoring

a. ONTARIO HYDRO, DESIGN AND DEVELOPMENT

a.1 Burchett, P.R., Brown, N.G. and Lyons, R.G. Ontario Hydro Research Centre Pump Test Complex Heat Transport Pump Test Loop Facility. Design and Development Division Report 80262.

Keywords: NUCLEAR / PUMP / TESTING

a.2 Burchett, P.R. and Sauve, R.G. Report on Pipe Vibration During Simulated Loss of Coolant Test on Bruce Pump at Kipling Ave. Design and Development Division Report 82194.

Keywords: NUCLEAR / PIPING / VIBRATION / TESTING

a.3 Burchett, P.R. and Ellis, P.J. Assessment of Primary Heat Transport System Pump Operation During A Loca. Design and Development Division Report 82197.

Keywords: NUCLEAR / PUMP / VIBRATION / TESTING

a.4 Stevens, T., Moffat, D.B and Lin, T.C. Frequency Response Analysis of PHT Pump Test Loop, Bruce GS 'A' and Pickering GS 'B' PHT Piping Systems Under Loss of Coolant Accident Condition. Design and Development Division Report 82236.

Keywords: NUCLEAR / PIPING / VIBRATION / TESTING

a.5 Lin, T.C Frequency Response Analysis of Bruce GS 'B' PHT Pump Discharge Piping System Under Loss of Coolant Accident Conditions. Design and Development Division Report 83260.

Keywords: NUCLEAR / PIPING / VIBRATION / ANALYSIS / TESTING

a.6 Fogarty, J.P. and Sauve, R.G. Frequency Response of Pickering NGS 'B' Primary Heat Transport Pump Test Loop. Design and Development Division Report 83292.

Keywords: NUCLEAR / PUMP / FLUID FORCES / TESTING

a.7 Lockett, J.C., Toong, T. and Burchett, P.R. Pickering NGS 'B' Assessment of Primary Heat Transport Pump Loss of Coolant Accident Tests. Design and Development Division Report 83328.

Keywords: NUCLEAR / PUMP / VIBRATION / TWO-PHASE / TESTING a.8 Stevens, T. and Manning, B W Frequency Response Analyses of Pickering GS 'B' PHT Piping Systems Under Loss of Coolant Accident Conditions. Design and Development Division Report 83410.

Keywords: NUCLEAR / PIPING / VIBRATION / ANALYSIS / TESTING

b. ONTARIO HYDRO, RESEARCH DIVISION

b.1 White, B.A. Bruce 'A' Heat Transport Pump Set Operation in a Steam Water Mixture. Ontario Hydro Research Division Report B82-21-K, 27/05/82.

Keywords: NUCLEAR / PUMP / PIPING / MOTOR / TWO-PHASE / TESTING

b.2 Perricone, F. Douglas Pt. G.S. Emergency Core Injection System Piping Vibration Analysis. Ontario Hydro Research Division Report 82-413-K, 22/10/82.

Keywords: NUCLEAR / PIPING / VIBRATION / ANALYSIS / TESTING

b.3 Liska, J.J. Review of Flow Induced Vibrations in Piping Networks. Ontario Hydro Research Division Report 81-502-K, 12/03/82.

Keywords: NUCLEAR / PIPING / VIBRATIONS / TWO-PHASE / ANALYSIS / TESTING

b.4 Jay, R.B. & Morgan, W.D. Pickering NGS 'B' - Moderator Pump Flow Tests. Ontario Hydro Research Division Report 81-314-K, 14/09/83. NOTE: Excluding Appendices.

Keywords: NUCLEAR / PUMP / CAVITATION / VIBRATION / TESTING

b.5 Perricone, F. BHWP Steam Condensate Return Pumpsets Vibration Analysis. Ontario Hydro Research Division Report 80-236-K, 18/07/80.

Keywords: PUMP / VIBRATION / ANALYSIS / TESTING

b.6 Koehler, H.P. Vibration Test on Motor End Windings: Bruce GS 'A' Primary Heat Transport Pump Rewound Motor. Ontario Hydro Research Division Report 80-426-H, 7/11/80.

Keywords: NUCLEAR / PUMPS / MOTOR / VIBRATION / TESTING / ANALYSIS

b.7 Tait, A.C Review of Work Progress and Programmes in 1981-82 at the Pump Test Complex. Ontario Hydro Research Division Report 82-77-H, 26/02/82.

Keywords: NUCLEAR / PUMP / TESTING

b.8 Koehler, H.P. Small Break Loss-of-Coolant Accident Test Bruce NGS 'A' PHT Pumpset Rotor, Hydraulic and Pipe Vibration. Ontario Hydro Research Division Report 82-239-H, 16/07/82.

Keywords: NUCLEAR / PUMP / PIPING / VIBRATION / TESTING

b.9 Boyd, J.M. Simulated Loss-of-Coolant Accident Tests Pickering NGS 'B' PHT Pumpset Rotor, Hydraulic and Pipe Vibration. Ontario Hydro Research Division Report 83-483-H, 12/12/83.

Keywords: NUCLEAR / PUMP / PIPING / VIBRATION / TESTING

b.10 Koehler, H.P. Simulated Loss-of-Coolant Accident Tests: Bruce NGS 'B' Pumpset: Rotor, Hydraulic and Pipe Vibration. Ontario Hydro Research Division Report B83-64-H, 14/10/83.

Keywords: NUCLEAR / PUMP / PIPING / VIBRATION / TESTING

b.11 Anderson, H.L. & Riddick, B. Flow-Induced Vibrations in Air-Filled Pipelines: A Progress Report. Ontario Hydro Research Division Report 82-25-K, 26/02/82.

Keywords: PIPING / VIBRATION / ANALYSIS / TESTING

b.12 Tait, A.C Review of Work Progress and Programmes in 1980/81 at the Pump Test Complex. Ontario Hydro Research Division Report 81-45-K, 2/03/81.

Keywords: PUMPS / TESTING

c. AECL

c.1 Metcalfe, R., Kittner, C.A. & Brown, G.W Effects of Pressure and Temperature Changes on End-Face Seal Performance. AECL, ASLE Transactions Volume 25, 3, pp. 361-371, 1981.

Keywords: NUCLEAR / SEALS / TESTING / ANALYSIS

c.2 Carlucci, L.N. Damping and Hydrodynamic Mass of a Cylinder in Simulated Two-Phase Flow AECL, Journal of Mechanical Design, Vol. 102, July 1980.

Keywords: NUCLEAR / VIBRATION / TWO-PHASE / TESTING / ANALYSIS

c.3 Meneley D.A. & Hancox, W T. LOCA Consequence Predictions in a CANDU-PHWR. AECL, Printed by International Atomic Energy Agency (Nuclear Power Fxperience Vol. 4), Vienna, 1983.

Keywords: NUCLEAR / TESTING

c.4 Earl, A.H. & Bastien, P. Bruce 'B' Heat Transport Pump Performance Report, Part 1, Assessment of the Mechanical Capability of the Pump to continue running following an LOCA. AECL, File 29-33122-200-001, 9/05/83.

Keywords: NUCLEAR / PUMP / TESTING / ANALYSIS / OPERATING

c.5 Bastien, P. & Kumar, A.N. Bruce 'B' Heat Transport Pump Performance Report. AECL, File 29-33122-200-00, 30/05/83.

Keywords: NUCLEAR / PUMP / ANALYSIS / OPERATING

c.6 Barclay F.W , Bean , D. and Nieman, R.F. Rama - A Computer Code for Analysis of Transient Two-Phase Flow and Heat Transfer in Pipe Networks. A.E.C L. Paper submitted to the Symposium on Simulation, Modelling and Decision in Energy Systems, June 1-2, 1978.

Keywords: NUCLEAR / TWO-PHASE / ANALYSIS / TESTING

c.7 Arrison, Bannerjee, et al. Blow Down of a Recirculating Loop with Heat Addition. A.E.C L. Paper 61. Institution of Mechanical Engineers, 1977.

Keywords: NUCLEAR / TWO-PHASE / TESTING

c.8 Heidrick, T.R., Hancox, W.T. & Nguyen, D. Centrifugal Pump Behaviour in Steady and Transient Steam-Water Flows. A.E.C L.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE /
ANALYSIS / TESTING

d. CANADA - OPERATIONS

d.1 Failure of Heat Transport Circulating Pump 3311-P6 Douglas Point G.S. Ontario Hydro Significant Event Report 75-20, 9/09/75.

Keywords: NUCLEAR / PUMP / OPERATING

d.2 Opening of a Heat Transport System Relief Valve While Operating at Full Power Pickering N.G.S. Ontario Hydro Significant Event Report 78-61, 16/05/78.

Keywords: NUCLEAR / OPERATING

d.3 Burchett, P. Review of Bruce 'A' Commissioning Event on Unit 2 (Accident Blowdown) August 16, 1975 and Comparison with PHT Pump Test Data. Ontario Hydro File 907N-33120-2P.

Keywords: NUCLEAR / PUMP / OPERATING

d.4 Langdon, R.A. Heat Transport Circulating Pump Seal Failure Pickering N.G.S. Ontario Hydro Significant Event Report 77-68, 21/04/77.

Keywords: NUCLEAR / PUMP / SEALS / OPERATING

d.5 Fulton, D. Failure of Heat Transport Pump No. 9 'O' Ring Seal Pickering N.G.S. Ontario Hydro Significant Event Report 79-18, 12/02/79.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING

d.6 Langdon, R.A. Heat Transport Circulating Pump # 6 Seal Change Pickering N.G.S. Ontario Hydro Significant Event Report 76-153, 28/10/76.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING

d.7 Heat Transport Pump # 6 Seal Failure and Reactor Trip During Unloading of Unit Douglas Point G.S. Ontario Hydro Memorandum to Production Superintendent, Douglas Point. G.S., 31/07/68.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING

d.8 Heat Transport Circulating Pump #10 Seal Failure Douglas Point G.S. Ontario Hydro Memorandum to Production Superintendent, Douglas Point G.S., 8/10/68.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING

e. U.S. PUMP /FLOW

e.1 Rothe, P.H. & Runstadler, P.W First - Order Pump Surge Behavior. ASME Paper 77-WA/FE-12.

Keywords: PUMP / PIPING / TWO-PHASE / FLUID FORCES / ANALYSIS

e.2 Kreps, D.A. & Kennedy, W G. C-E/EPRI Tow-Phase Primary Pump Performance Program. Proc. of Topical Meeting on Thermal Reactor Safety. Sun Valley, Idaho, Jule 31 - Aug 4, 1977.

Keywords: PUMP / CENTRIFUGAL / TWO-PHASE / TESTING

e.3 Marr, A. M-C., Phelps, R.L. & Katz B Loss of Component Cooling Water Capability of a PWE Reactor Coolant Pump. ASME Paper

Keywords: NUCLEAR / PUMP / TESTING

e.4 Sparks, C.R. & Wachel, S.C. Pulsations in Centrifugal Pump & Piping Systems. Proc. of the Turbomachinery Symposium, 1977.

Keywords: PUMP / CENTRIFUGAL / PIPING

e.5 Donelson, R.K., Zarlenga, B A. & Colwill, W.H. Pump Cavitation Prediction using Three-Dimensional Flow Analysis. Allis - Chalmers Fluid Products Report.

Keywords: PUMP / TWO-PHASE / ANALYSIS

e.6 Chen, T-H. & Mopro, S.M Transient Two-Phase Performance of Loft Reactor Coolant Pumps. ASME Winter Annual Meeting, Nov 13-18, 1983.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS

e.7 Wilson, Chan, et al. Performance Prediction of Centrifugal Pumps in Two-Phase Flow. Proc. of the Joint Symposium on Design and Operation of Fluid Machinery,. Vol. II, Colo. State Univ. Jun 12-14, 1978.

Keywords: PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS e.8 Runstadler P.W & Dolan, F.X. Two-Phase Flow Pump Data for a Scale Model NSSS Pump. Polyphase Flow in Turbomachinery, Winter Annual Meeting of the ASME, San Francisco California, Dec 10-15, 1978.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS / TESTING

e.9 Patel, B R. & Runstadler, P.W Investigations into the Two-Phase Flow Behavior of Centrifugal Pumps. Polyphase Flow in Turbomachinery, Winter Annual Meeting of the ASME, San Francisco California, Dec 10-15, 1978.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS / TESTING

e.10 Hunter, J.A. & Harris, P.A. Performance of Small Nuclear Reator Primary Coolant Pumps Under Blowdown Conditions. Proc. of the Joint Symposium on Design & Operations of Fluid Machinery, Vol. 1, Colo. State Univ. June 12-14, 1978.

Keywords: NUCLEAR / PUMP / TWO-PHASE / ANALYSIS / TESTING

e.11 Penzes, L.E. Theory of Pump Induced Pulsating Coolant Pressure in Pressurized Water Reactors. Transactions of the 4th International Conference on Structural Mechanics in Reactor Technology, Vol. E, San Francisco California, 15-19 August, 1977.

Keywords: NUCLEAR / PUMP / ANALYSIS

e.12 Olson, D.J. Experimental Data Report for Single and Two-Phase Steady State Tests of the 1 1/2 Loop Mod-1 Semiscale System Pump. U.S. Atomic Energy Commission, May 1974.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE / TESTING

e.13 Johnsen, Cozzuol, et al. Experimental Evaluation of the Effect of Primary Coolant Pump Operation During Small Break LOCA. Small Break LOCA Analysis in LWR's Conference Papers, 25-27 August, 1981.

Keywords: NUCLEAR / PUMP / TESTING

e.14 Mikielewicz Chan, et al. A Method for Correlating the Characteristics of Centrifugal Pumps in Two-Phase Flow. ASME Paper 76-WA/FE-29.

Keywords: PUMP / TWO-PHASE / TESTING / ANALYSIS e.15 Chamien, D. and Acosta, A.J. Dynamic Forces on a Whirling Centrifugal Rotor. Paper Published in Proceedings of the Sixth Conference on Fluid Machinery, Vol. 1. Budapest 1979.

Keywords: PUMP / CENTRIFUGAL / CAVITATION / FLUID FORCES / VIBRATION / ANALYSIS

e.16 Rothe, P.H., Runstadler, P.W and Doan, F K. Pump Surge due to Two-Phase Flow. New York: A.S.M E. Presented at the Winter Annual Meeting of the A.S.M E. San Francisco, California, Dec 10-15, 1978.

Keywords: PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS / TESTING

e.17 Lee, L.C. & Chandra, S. Pump Induced Fluctuating Pressure in a Reactor Coolant Pipe. Int. Journal of Pressure Vessels & Piping, 1980.

Keywords: NUCLEAR / PUMP / PIPING / VIBRATION / OPERATING

e.18 VonNimitz W.W Low Frequency Vibrations at Centrifugal Plants. Proc. of 4th Turbomachinery Symposium.

Keywords: PUMP / CENTRIFUGAL / VIBRATION / OPERATING

e.19 Lee, L. and Chandra, S. Pump-Induced Fluctuating Pressure in a Reactor Coolant Pipe. Combustion Engineering Inc, Windsor, Conn. Interantional Journal of Pres. Ves & Piping, Applied Science Publishers Ltd, England, November 1979.

Keywords: NUCLEAR / PUMP / PIPING / VIBRATION / OPERATING

f. U.S. - FLOW (ONLY)

f.1 Fisher, R.K. & Donelson, R.K. Characteristics of Francis Turbines Operating in Cavitating Regimes. Research & Development Department, Allis-Chalmers Fluid Products Company, York, Pennsylvania.

Keywords: TWO-PHASE / ANALYSIS

g. U.S. PUMP/MECHANICAL/VIBRATION/SEALS/MOTORS

g.1 Nuclear Energy Agency. Reactor Coolant Pump Seal Failure. O.E.C.D., PNO-IV-80-12, 12/05/80.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING

g.2 Nuclear Energy Agency. Reactor Coolant Pump Seal Failure (Supplemental Information). O.E.C.D., PNO-IV-80-21A, 13/05/80.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING

g.3 Nuclear Energy Agency. Reactor Coolant Pump Seal Failure (Supplemental Information). O.E.C D., PNO-IV-80-21B, 14/05/80.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING

g.4 Nuclear Energy Agency. Reactor Coolant Pump Seal Failure (Supplemental Information). O.E.C D., PNO-IV-80-21C, 16/05/80.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING

g.5 Lanning, W.D. & Brown, E.J. Pressure Boundary Degradation due to Pump Seal Failure at Arkansas Nuclear One. N.R.C., Ninth Meeting of the Committee on Safety of Nuclear Installations, Paris, France, Nov. 3-4, 1981.

Keywords: NUCLEAR / PUMP / SEAL / OPERATING / ANALYSIS

g.6 Miller, J.G. Letter from J.G. Miller, Metropolitan Edison Co., to Dr. D.F. Knoth, A.E.C , Docket # 50-289, 9/11/73.

g.7 Sosa, F & Baker, E.W. Reducing Casing Vibrations on Large Vertical Circulating Water Pumps. A.S.M.E. Paper 77-Fe28.

Keywords: PUMPS / VIBRATIONS / ANALYSIS

g.8 Rangwala, A.S. & Stadler, E.L. Mechanical & Thermal Stresses in the Volute of a Pump Casing. A.S.M E. Paper 78-PVP-45.

Keywords: PUMP / ANALYSIS / MECHANICAL DESIGN

g.9 Adams, M.L., Makay E. & Dimmer, J.P. Feed Pump Vibration Research - Interstage Fluid Annuli. A.S.M.E. Paper, 80-JPGC-PWR-20.

Keywords: NUCLEAR / PUMP / VIBRATION / ANALYSIS

h. U.S. - PIPING GENERAL

h.1 McGeorge, R. & Swec, L.F. The Steam Hammer Problem: Dynamic Shock Loading of Critical Reactor Piping Systems. North-Holland Publishing Company, Nuclear Engineering & Design 32 (1975) pp. 121-128.

Keywords: NUCLEAR / PIPING / VIBRATION / TWO-PHASE / ANALYSIS

i. U.S. - VIBRATION (GENERAL)

i.1 Kovats, A.B. Vibration of Vertical Pumps. ASME Paper 61-HYD-10. Presented at the ASME-EIC Hydraulic Conference, Montreal, Canada, May 7-10, 1961.

Keywords: PUMP / VIBRATION

i.2 Kassawars, R.P. & Bushman, F H. Dynamic Analysis of Reactor Coolant Systems under LOCA Conditions. North-Holland Publishing Company, Transactions of the 4th International Conference on Structural Mechanics in Reactor Technology, Vol. F, San Francisco, California. 15-19 Aug., 1977.

Keywords: NUCLEAR / PIPING / MECHANICAL DESIGN / VIBRATION / ANALYSIS

i.3 Stenson, K.A. Perturbation Method of Structural Design Relevant to Holographic Vibration Analysis. AIAA Journal Vol. 13, No. 4, April 1975.

Keywords: VIBRATION / MONITORING

j. U.S. - SEISMIC

j.1 McNutt, L.C. Nonlinear Transient and Spectrum Response Seismic Analysis of a Reactor Coolant Pump. ASME Paper 78-PVP-36.

Keywords: NUCLEAR / PUMP / SIESMIC / ANALYSIS

j.2 McDonald, C.K. Seismic Qualifications by Analysis of Nuclear Power Plant Mechanical Components. University of Alabama.

Keywords: NUCLEAR / SIESMIC / ANALYSIS

j.3 Villasor, A.P. Response of a Reactor Coolant Pump to Time History Seismic Displacements. North-Holland Publishing Company, Transactions of the 6th International Conference on Structural Mechanics in Reactor Technology Paris, France, 1981.

Keywords: NUCLEAR / PUMP / SIESMIC

j.4 Villasor, A.P. Seismic Analysis of a Reactor Coolant Pump by the Response Spectrum Method. North-Holland Publishing Company, Nuclear Engineering and Design 38(1976) pp. 527-542.

Keywords: NUCLEAR / PUMP / SIESMIC / ANALYSIS

j.5 McDonald, C K. Seismic Analsyis of Long Column Vertical Pumps. ASME Paper 74-NE-2, Presented at the Pressure Vessels and Piping Conference, Miami Beach, Florida, June 24-28, 1974.

Keywords: PUMP / SIESMIC / ANALYSIS

k. INTERNATIONAL-OPERATING

k.1 Fagone, S., Letter from S. Ragone, Vepco, to A. Giambusso, AEC. Docket # 50180-199.

Keywords: NUCLEAR / PUMP / VIBRATION / OPERATING

k.2 Incident Reporting System. CAORSO BWR. NEA/IRS - 109, 1981.

Keywords: NUCLEAR / PUMP / MOTOR / ELECTRICAL / VIBRATION / OPERATING

k.3 Incident Report - 19, Dampierre,. Nuclear Energy Agency, Feb 1980.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / VIBRATION / OPERATION / SIESMIC

k.4 Incident Reporting System, Reactor Feed Pump Trip. OECD Nuclear Energy Agency.

Keywords: NUCLEAR / PUMP / TWO-PHASE / VIBRATION / OPERATION

k.5 Analysis of the Three Mile Island - Unit 2 Accident. Technical Summary, NSAC - 80-1, Nuclear Safety Analysis Centre, EPRI, Palo Alto, California, Revised March 1980.

1. FRANCE

1.1 Corniglion, J. & Albano, G. Theoretical and Experimental Study of Flow Through Centrifugal Pumps. Proc Joint Symp. on Design and Operation of Fluid Machinery Colorado State Univ. June 12-14, 1978.

Keywords: PUMP / CENTRIFUGAL / FLUID FORCES / ANALYSIS / TESTING

1.2 Gibert, R.J. & Axisa, F. Flow Induced Vibrations of Piping. Transactions of the 4th International Conference on Structural Mechanics in Reactor Technology, Vol. F, San Francisco California 15-19 August, 1977.

Keywords: NUCLEAR / PIPING / VIBRATION / ANALYSIS / TESTING

1.3 Grison, P. & Lauro, J.F. Experimental and Theoretical Investigations about Two-Phase Critical Flow Through a Pump. Polyphase Flow in Turbomachinery ASME, San Francisco, Dec 1978.

Keyw[illegible]: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE / FLUID FORCES / ANALYSIS / TESTING

1.4 Grison, P. & Lauro, J.F. Two-Phase Performance Characteristics of a PWR Primary Pump Under LOCA Conditions. Heat and Fluid Flow in Water Reactor Safety, I-Mech-E Conference Publications 1977-8.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS / TESTING

1.5 Gibert, R.J. & Axisa, F Flow Induced Vibrations of Piping. Transactions of the 4th International Conference on Structural Mechanics in Reactor Technology, Vol.F., San Francisco California, 15-19 August, 1978.

Keywords: NUCLEAR / PIPING / FLUID FORCES / ANALYSIS / TESTING

1.6 Nuclear Energy Agency Report. Critical Flow Modelling in Nuclear Safety. A State-of-the-Art Report by a Group of Experts of the NEA Committee on the Safety of Nuclear Installations, June 1982.

Keywords: NUCLEAR / TWO-PHASE / ANALYSIS

1.7 Department Transferts Thermiques et Aerodynamique, L'Installation Diphasique a Rafales (Indira). Electricite de France, 8/5/79.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS / TESTING

1,8 Grison, P. Letter from P. Grison, to P. Burchett, Design & Development Division, Ont. Hydro, 28/3/80.

m. GERMANY

m.1 Schafer, C & Simon, C. Experimental Investigation of the Cavitation Behavior of Boiling Water Reactor Coolant Pumps. Kraftwerk Union, Germany.

Keywords: NUCLEAR / PUMP / TWO-PHASE / TESTING

m.2 Olma, B J. Structurals Analysis and Incipient Failure Detection of Primary Circuit Components Based on Correlation - Analysis and Finite Elements Models. Transactions of the 4th International Conference on Structural Mechanics in Reactor Technology Vol.F, Edited by Thomas A. Jaeger & Bruno A. Boley, International Association for Structural Mechanics in Reactor Technology, 1977.

Keywords: REF

m.3 Kastner, W & Seeberger G.J. Pump Behavior and its Impact on a Loss-of-Coolant Accident in a Pressurized Water Reactor. Nuclear Technology Vol. 60, Feb 1983, American Nuclear Society, 1983.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS

n. JAPAN

n.1 Uchida, N., Imaichi, K., Shirai, T., Radial Force on the Impeller of a Centrifugal Pump. Bulletin of JSME Vol. 14, No.76, 1971.

Keywords: PUMP / CENTRIFUGAL / TWO-PHASE / FLUID FORCE / TESTING

n.2 Okamura, T. & Mivashiro, H., Cavitation in Centrifugal Pumps Operating at Low Capacities. Polyphase Flow in Turbomachinery, ASME Aug 10 1979.

Keywords: PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS / TESTING

n.3 Murakami, M & Minemura, K. Flow of Air Bubbles in Centrifugal Impellers and its Effect on Pump Performance. 6TH Australasian Hydraulics and Fluid Mechanics Conference, Institution of Engineers, Australia National Conference Publication No. 77/12.

Keywords: PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS / TESTING

n.4 Michiyuki, T., Analysis & Design of Centrifugal Pumps Considering Rotor Dynamics. Second International Conference on Vibrations in Rotating Machinery (1980). I Mech E Conference Publications 1980-4.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / VIBRATION / FLUID FORCES / ANALYSIS

n.5 Yamagugmi, Y. & Miura, S. Characteristics of Radial Force Fluctuations on Runners of Francis - Type Pump Turbines. Bulletin of JSME vol.24, no. 195, Sept. 81.

Keywords: PUMP / MECH. DESIGN / VIBRATION / ANALYSIS

n.6 Lino, T., Sato, H. & Miyashiro, H. Hydraulic Axial Thrust in Multistage Centrifugal Pumps. Transaction of the ASME, Vol 102, Mar, 1980.

Keywords: PUMP / CENTRIFUGAL / MECH DESIGN / ANALYSIS

n.7 Hiramatsu, T., Komura, Y. Yano, S. Analysis of Two-Phase Flow Induced Vibrations in Piping Systems. Flow-Induced Vibration of Circular Cylindrical Structures 1982, PVP-Vol 63.

Keywords: NUCLEAR / PIPING / TWO PHASE / FLUID FORCES / VIBRATION / ANALYSIS

o. ITALY

o.1 Marenco, S.G. & Vergani, L. "Elastic Solution" For Vibration Trouble on Verticle Pumps. L'Energia Elettrica Supp. 10-1982.

Keywords: PUMP / VIBRATION / ANALYSIS / TESTING

p. U.K.

p.1 Brown, R.D. Vibration Phenomena in Boiler Feed Pumps Originating from Fluid Forces. L'Engeria Eletrica Vol. LIX, Supp. 10-1982.

Keywords: PUMP / CENTRIFUGAL / FLUID FORCES / VIBRATION / ANALYSIS

p.2 Brown, R.D. Vibration Phenomena in Large Centrifugal Pumps. Vibrations and Noise in Pump, Fan, and Compressor Installations, Conference Sponsored by the Fluid Machinery Group of the Institution of Mechanical Engineers, University of Southampton, 16-18 Sept. 1975.

Keywords: PUMP / CENTRIFUGAL / FLUID FORCES / VIBRATION / ANALYSIS

p.3 Black, H.F. Calculation of Forces Whirling and Stability of Centrifugal Pump Rotor Systems. ASME Paper 73-DET131.*

Keywords: PUMP / CENTRIFUGAL / FLUID FORCES / ANALYSIS

p.4 Brown, R.D. & France, D. Vibration Response of Large Boiler Feed Pumps. Pumps and Pumping Systems for Liquids in Single or Multiphase Flow, Ulrico Moepli Ed. 1974.

Keywords: PUMP / CENTRIFUGAL / FLUID FORCES / VIBRATION / ANALYSIS

p.5 Black H.F. Lateral Stability and Vibrations of High Speed Centrifugal Pump Rotors. Editor F.I. Niordson, Dynamics of Rotors, 1974.

Keywords: PUMP / CENTRIFUGAL / VIBRATION / FLUID FORCES / ANALYSIS

p.6 Black, H.F. Effects of Hydraulic Forces on Annular Pressure Seals on the Vibrations of Centrifugal Pump Rotors. Journal of Mechanical Engineering Science, Vol. 11 No 2, 1969.

Keywords: PUMP / CENTRIFUGAL / SEAL / FLUID FORCE / VIBRATION

---

* including a Discussion of the paper by E. Mackay Consulting Engineers, Energy Research & Consultants Co., Morrisville, Pennylvannia.

q. EASTERN BLOCK COUNTRIES

q.1 Sebestven, Gy., Rizk, M & Szabo, A. Some Properties from Cavitation - Vibration Tests on Centrifugal Pumps. Proc. Joint Symposium on Design & Operation of Fluid Machinery, Vol. II, Colorado State University, June 10-14, 1978.

Keywords: PUMP / CENTRIFUGAL / TWO-PHASE / VIBRATION / TESTING

q.2 Mikielewicz, Wilson, et al. A Method for Correlating the Characteristics of Centrifugal Pumps in Two-Phase Flow. ASME, Paper 76-WA/FE-29.

Keywords: REF

q.3 Kirejczyk, J. Cavitation Signs in Rotodynamic Pump. Second International Conference on Cavitation, I Mech E Conference Publications 1983-8.

Keywords: PUMP / TWO-PHASE / ANALYSIS / TESTING

r. MISCELLANEOUS COUNTRIES

r.1 Keskinen, R.P. Hydro Elastic Piping Vibration Analysis by Transfer Matrices. Int. J. Pres Ves. & Piping 9(1981) pp. 263-283.

Keywords: PIPING / FLUID FORCES / VIBRATION / ANALYSIS

r.2 Barp, B Dynamic Behavior of Large Pump Turbine Rotors. Water Power and Dam Construction, Jan 1976, pp. 48-51.

Keywords: PUMP / FLUID FORCES / VIBRATION / ANALYSIS

s. COMPUTER CODES

s.1 Fabic, S. Computer Codes in Water Reactor Safety : Problems in Modelling of Loss-of-Coolant Accident. Heat and Fluid Flow in Water Reactor Safety I MECH E Conference, Publications, 1977-8.

Keywords: NUCLEAR / TESTING

s.2 MacNeal, R.H., Citerly, R. & Chargin, M A New Method for Analizing Fluid-Structure Interaction using MSC/NASTRAN. Transactions of the 4th International Conference on Structural Mechanics in Reactor Technology, Vol. B, San Francisco, California, 15-19 August, 1977.

Keywords: NUCLEAR / FLUID FORCES / TESTING / ANALYZING

s.3 Belytschko, T. Methods and Programs for Analysis of Fluid Structure Systems. Nuclear Engineering & Design 42 (1977) pp. 41-52.

Keywords: NUCLEAR / PIPING

t. BACKGROUND REFERENCES

t.1 Wambsganss, M W Second Order Effects as Related to Critical Coolant Flow Velocities and Reactor Parallel Plate Fuel Assemblies. Nuclear Engineering & Design, Vol. 5, No. 3, pp. 268-276, 1967.

Keywords: NUCLEAR / VIBRATION

t.2 Esmleman, R.L. Vibration Standards, Shock and Vibrations Handbook Second Edition. Edt. C.M Madrix & C.E. Crede, McGraw-Hill Book Co.

Keywords: VIBRATION / ANALYSIS

t.3 Parmakian, J. Water Hammer. Pump Handbook. Edt. Karassik, Fraser, et al. McGraw-Hill Book Co.

Keywords: PUMP / PIPING / TWO-PHASE / ANALYSIS

t.4 International Standards Organization (ISO). Mechanical Vibration of Machines with Operating Speeds from 10 to 200 rev/s - Basis for Specifying Evaluation Standards. International Organization for Standardization, 11/01/74.

Keywords: PUMP / VIBRATION / ANALYSIS

t.5 Pickering 'B' Design Manual. Heat Transport System Main Circuit. NK30-33100, Dec. 83.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / SEALS / MECHANICAL

t.6 Pickering 'B' Safety Report. Reactor Coolant System and Connected Systems, Chapt. 1 Section 5.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / PIPING / TWO-PHASE / ANALYSIS

t.7 Bruce 'B' Design Description. Primary Heat Transport System. Sci 33100/63310 Apr. 82.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / SEALS / MECHANICAL

t.8 Bruce 'B' Safety report. Reactor Process Systems. Chap. 2, Section 4.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / PIPING / TWO-PHASE / ANALYSIS t.9 Bruce 'B' Safety Report. Reactor Process Systems. Chap. 3, Sections 3.5 - 3.7

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / PIPING / TWO-PHASE / ANALYSIS

t.10 Darlington G.S. 'A' Design Description. Main Heat Transport Circuit. Sci 33100, Nov. 1978

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / SEALS / MECHANICAL

t.11 Darlington G.S. 'A' Safety Report. Reactor Process Systems. Chapt. 2, Section 4.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / PIPING / TWO-PHASE / ANALYSIS

t.12 Pt. Lepreau Design Manual. Main Heat Transport System. DM - 33100, Oct. 78.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / SEALS / MECHANICAL

t.13 Gentilly - 2. Safety Report Vol. 1. A.E.C L Rev 81.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / PIPING / TWO-PHASE / ANALYSIS

t.14 Gentilly - 2, Design Specification, Heat Transport Pumps. TS-66-33122-1, AECL, Rev 1, 1973.

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / SEALS / MECHANICAL

t.15 Gentilly - 2, Design Specification, Heat Transport Pump Motors. TS-66-33121-1, AECL, Rev 1, 1973

Keywords: NUCLEAR / PUMP / CENTRIFUGAL / SEALS / MECHANICAL

t.16 Banerjee, S., Heidrick, T.R., Rhodes, E. Development and Calibration of Instruments for Measurements in Transient Two-Phase Flow. Paper presented at the second C.S.N.I. (Committee on the Safety of Nuclear Installations) Specialist Meeting on Transient Two-Phase Flow Paris, June 1978.

Keywords: NUCLEAR / TWO-PHASE / MONITORING

t.17 Banerjee, S., Heidrick, T.R., Saltvold, J.R. Measurenent of Void Fraction and Mass Velocity in Transient Two-Phase Flow. . Paper presented at the specialists meeting on Transient Two-Phase Flow. Toronto, Ont. August 3-4, 1976.

Keywords: NUCLEAR / TWO-PHASE / MONITORING

t.18 Brennen, Runstadler, et al. Polyphase Flow in Turbomachinery. New York: A.S.M.E. Presented at the winter annual meeting of the A.S.M.E. San Francisco, California Dec 10-15, 1978.

Keywords: NUCLEAR / PUMPS / CENTRIFUGAL / TWO-PHASE / VIBRATION

t.19 Tong, L.S. and Wieshan, J. Thermal Analysis of Pressurized Water Reactors. Second Edition. Illinois: American Nuclear Society, 1979.

Keywords: NUCLEAR / ANALYSIS

t.20 Wylie, E.B. and Streeter, V.L. Fluid Transients. New York: McGraw - Hill, Inc. 1978

Keywords: FLUID FORCES

t.21 Wilson, Chan, et al. Analytical Models and Experimental Studies of Centrifugal Pump Performance in Two-Phase Flow. EPRI NP-677 Electric Power and Research Institute, Palo Alto, California. May 1979

Keywords: NULCEAR / PUMP / CENTRIFUGAL / TWO-PHASE / ANALYSIS / TESTING

t.22 Rubin, S. Longitudinal Instability of Liquid Rockets Due to Propulsion Feedback (POGO). Journal of Spacecraft and Rockets, 3, No. 8, 1966.

APPENDIX E

Telex Messages

DE Telex....................................page E1

Addresses...................................page E2

ISPRA Response..............................page E4

Worthington Response........................page E5

Kraftwerk Union Response....................page E6

FEB 24/84
DE MSG NO 00024

WE ARE WORKING WITH ATOMIC ENERGY CONTROL BOARD OF CANADA ON AN INTERNATIONAL LITERATURE SURVEY. SUBJECT: REACTOR COOLANT PUMP VIBRATION DUE TO TWO PHASE/ ACCIDENT CONDITIONS.

PARTICULARLY INTERESTED IN (1) EXPERIMENTS, DATA OR INCIDENTS CAUSING VIBRATION (2) NATURE OF FLUID FORCING FUNCTION (3) EFFECT OF PIPING COUPLED TO PUMP (4) EMPIRICAL METHODS DEVELOPED WHICH CORRELATE PUMP-PIPING VIBRATION TO SYSTEM PARAMETERS.

DO YOU HAVE RELEVANT INFORMATION WHICH WE MIGHT OBTAIN? WHO WOULD BE CONTACTS (NAMES,POSITIONS, ADDRESS, TELEPHONE AND TELEX) WE MAY ARRANGE A TRIP TO EUROPE.

IF NO RELEVANT INFORMATION, DO YOU KNOW OF OTHER EUROPEAN CONTACTS?

WOULD APPRECIATE RESPONSE BY 02 MAR 84.

PLEASE ADDRESS TO : B. GUTHRIE
DAVIS ENGINEERING
1481 CYRVILLE RD.
OTTAWA, ONTARIO
CANADA, K1B 3L7
TELEPHONE (613) 748 5500
TELEX 053 3862

- L. Piplies
  Ispra Establishment
  Div. of Ht. Transfer and Fluid Dynamics
  I - 21020 Ispra (Varese), Itlay

  Telex: 380042 Eur I

- S.G. Marenco
  Worthington Nord SpA
  20033 Desio (Milan), Italy

  Telex: 333023 Wormil I

- H. Schafer
  Kraftwerk Union - Karlstein Labs
  D-8520 Erlangen 2
  Federal Republic of Germany

  Telex: 6 29290

- P. Grison
  Electricite de France
  Direction des Etudes et Recherches
  Chatou, France

  Telex: 696395 F EDFDER

- R.J. Gibert

  Dept. des Etudes Mecaniques et Thermiques
  Centre d'Etudes Nuclearie de Saclay
  Gif-sur-Yvette, France

  Telex 690 641 F Energat Saclay

- T. Mumford

  Weir Pumps Ltd.
  Glasgow U.K

  Telex 77161/2

◆

DAVIS ENG OTT

380042 EUR I

TELEX NR. 1264 1.03.84 16.28 H.

TO B. GUTHRIE
DAVIS ENGINEERING
OTTAWA - ONTARIO

1613/135/84/BD - PS/BS

REF.: YOUR TELEX 24.2.1984

BORPUMP THE INFLUENCE OF TWO-PHASE FLOW CONDITIONS ON HEAD AND TORQUE WAS INVESTIGATED. NO VIBRATION TESTS HAVE BEEN PERFORMED.

(L. PIPLIES)
LOBI PROJECT
J.R.P. ISPRA ESTABLISHMENT - ITALY

◆

DAVIS ENG OTT

380042 EUR I.........

+
DAVIS ENG OTT

332137 WORMIL I

COMPUTER MESSAGE DON'T DIGIT

B. GUTHRIE

AUT.TX DONT INTRPT (04769-   0) TX. 04769

ATTENTION: B. GUTHRIE - DAVIS ENGINEERING

SUBJECT: YOUR TELEX OF FEBRUARY 24TH
REACTOR COOLANT PUMP VIBRATION DUE TO TWO PHASE/ACCIDENT CONDITIONS

I HAVEN'T RELEVANT INFORMATION REGARDING QUESTIONS INDICATED IN YOUR ABOVE TELEX.
HOWEVER I WOULD LIKE TO INFORM YOU THAT EXPERIMENTAL RESEARCHS ON THE SUBJECT ARE SCHEDULED STARTIN FROM THE NEXT YEAR IN TWO SEPARATE TEST LOOPS AT: TURIN POLYTECHNIC AND CISE LABORATORIES

IF FURTHER DETAILED INFO ARE NEEDED, PLS CONTACT:
- ING. MAZZOCCHI C/O CISE LABORATORIES - VIA REGGIO EMILIA 39
  SEGRATE-MILAN PHONE 02-2133241 TX 311643 CISE I
- INGG. PANELLA, DEL TIN C/O ENERGY DEPT. - TURIN POLYTECHNIC -
  CORSO DUCA DEGLI ABRUZZI 24 - TURIN PHONE 011 510447
  TX. 22646 POLITO

BEST REGARDS

G. MARENCO - WORTHINGTON DESIO (ITALY)
NNNN
332137 WORMIL I+
DAVIS ENG OTT
MMMM+
DAVIS ENG OTT

M•
DAVIS ENG OTT

4184210 KWU D

FS-NR. 535   28.02.84   11.00 H MR. SCHAEFER/ R 541/ STG

C/O MR. B.GUTHRIE, DAVIES ENGINEERING, 1481 CYRVILLE RD.
OTTAWA, ONTARIO /CANADA

V.KWU - KARLSTEIN , R 541, HR.SCHAEFER, TEL.06188/70274

YOUR INQUIRY NR. 00024 FROM FEB. 24/84

TO REPLY YOUR INQUIRY FROM FEB. 24/84 WE CAN TELL YOU THAT WE HAVE GOT SOME EXPERIMENTAL EXPERIENCES WITH REACTOR COOLANT PUMP VIBRATION DUE TO TWO PHASE FLOW CONDITIONS. WE GOT THE INFORMATION BY TESTING SOME INTERNAL RECIRCULATION PUMPS OF OUR BWR PLANTS ON OUR FULL SCALE PUMP TEST FACILITY UNDER CAVITATION CONDITIONS WHERE PUMP VIBRATION (SHAFT,IMPELLER, MOTOR) OCCURES DUE TO TWO PHASE FLOW.

THE RESULTS OF THESE TESTS WERE PUBLISHED ON ''JAHRESTAGUNG KERNTECHNIK 1978''* AND ON THE SIXTH CONFERENCE ON FLUID MACHINERY 1979 IN BUDAPEST. PLEASE TELL US IF YOU ARE INTERESTED IN THE COMPACT OF OUR LECTURE.

CONTACTS: KRAFTWERK UNION AG
C/O MR. H.R.SCHAEFER
ABT. R 541
SELIGENSTAEDTERSTR.
D - 8757 KARLSTEIN/MAIN

YOURS SINCERLY

H.R. SCHAEFER

KWU/KARLSTEIN

•
DAVIS EOD OTT
V
4184210 KWU D•
DAVIS ENG OTT

4184210 KWU D

* Reference (29)